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Bethesda, Maryland 20084

U.S. COAST GUARD 270-FT MEDIUM ENDURANCE CLASS CUTTER FIN STABILIZER PERFORMANCE

by

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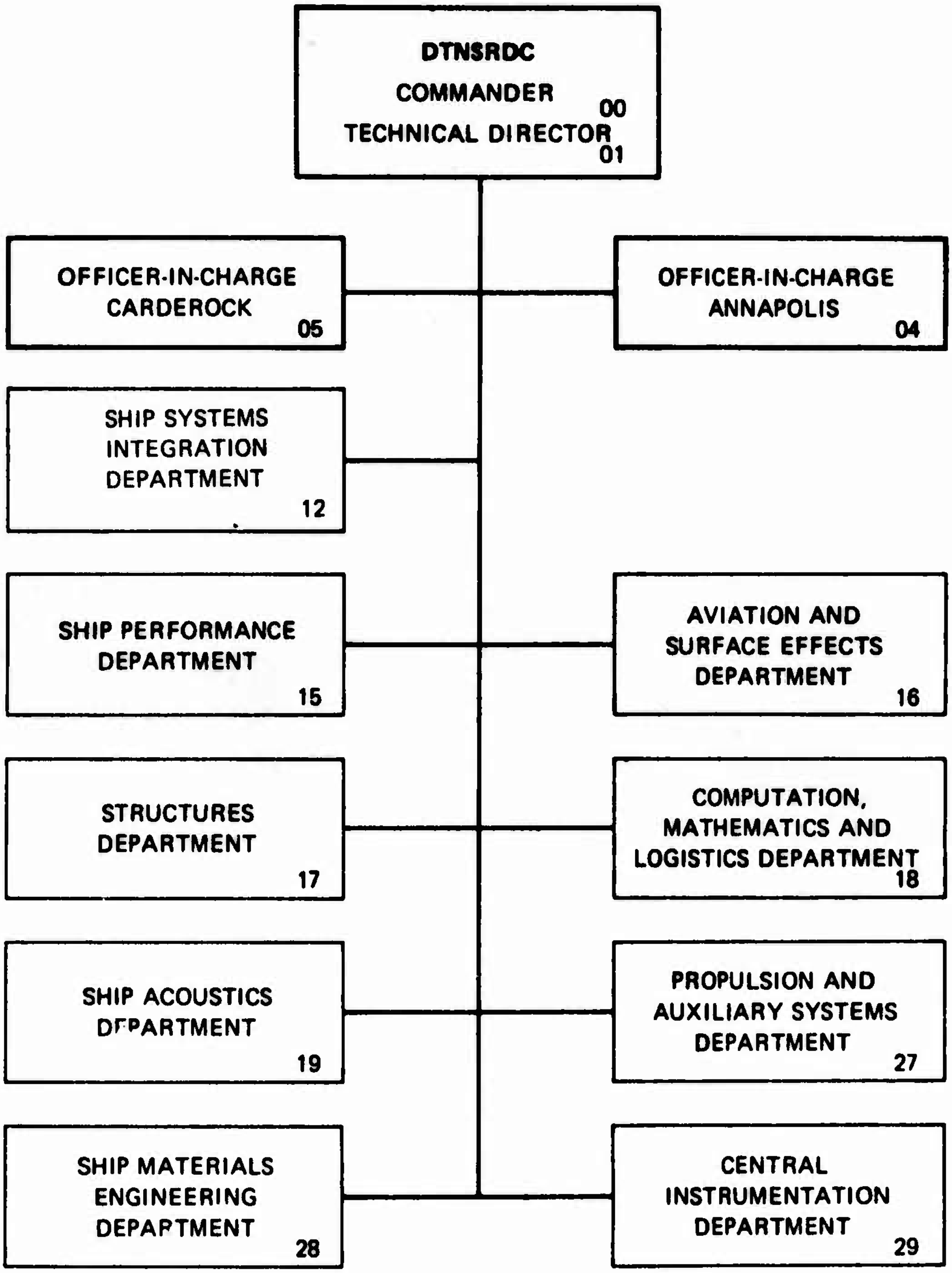

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As part of the seakeeping trials conducted on board the United States Coast Guard Cutter BEAR (WMEC 901), anti-roll fin stabilizer system performance was to be evaluated. The fin system as installed and operated aboard the USCGC BEAR, however, exhibited a number of deficiencies which prevented an accurate assessment of its roll reduction capability. Among the problems which were identified were intermittent excessive travel of the port fin; improper speed input into the controller; incorrect operation of the MANUAL versus AUTOMATIC GAIN modes by the crew; and a defective roll angle sensor which degraded the control algorithm, particularly in quartering seas.

It was further discovered by subsequent data reduction that highly desirable alterations should be made to increase the size of the bilge keels and fins to obtain optimum roll damping. Such changes are considered crucial in order to overcome the excessive degradations in crew performance (i.e., fatigue and motion-induced interruptions) caused by the large vertical accelerations in the ship's work areas.

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ABSTRACT

As part of the seakeeping trials conducted on board the United States Coast Guard Cutter BEAR (WMEC 901), anti-roll fin stabilizer system performance was to be evaluated. The fin system as installed and operated aboard the USCGC BEAR, however, exhibited a number of deficiencies which prevented an accurate assessment of its roll reduction capability. Among the problems which were identified were intermittent excessive travel of the port fin; improper speed input into the controller; incorrect operation of the MANUAL versus AUTOMATIC GAIN modes by the crew; and a defective roll angle sensor which degraded the control algorithm, particularly in quartering seas.

It was further discovered by subsequent data reduction that highly desirable alterations should be made to increase the size of the bilge keels and fins to obtain optimum roll damping. Such changes are considered crucial in order to overcome the excessive degradations in crew performance (i.e., fatigue and motion-induced interruptions) caused by the large vertical accelerations in the ship's work areas.

In addition to fin enlargement, corrective actions are recommended to bring all BEAR-class fin systems up to full capacity. These include step-by-step instructions for the crew in the check-out and operation procedures, and a reduction in the maximum fin angle limit.

ADMINISTRATIVE INFORMATION

This investigation was authorized by the United States Coast Guard under MIPR's Z70099-4-00758 and Z70098-5-32053, identified at David Taylor Naval Ship Research and Development Center as Work Units 1561-047 and 1561-049, respectively.

TRIAL OBJECTIVES

The objectives of the roll fin performance element of the BEAR seakeeping trial was to:

1. Document the performance of the fin system as installed on the ship;
2. Define the system deficiencies as noted and distinguish between ship set and class system problems;
3. Collect data to develop system performance enhancements.

The primary objectives were the definition of the roll reduction performance of the installed fin stabilizer system and the related system deficiencies. This latter component, namely the identification and correction of all fin system malfunctions, has proven to be difficult. The correction of some of the system malfunctions has not been completed as part of the current work since they were not identified until after the completion of the trial and are beyond the scope of the current trial effort.

One aspect of the primary objectives was the identification of the impact of fin activity on the vertical responses of the ship, i.e., pitch and vertical acceleration at various positions on the ship. It was identified prior to the seakeeping/fin performance sea trials, and later confirmed through analysis of the trials' results, that vertical motions were contributing to serious crew performance degradation.[1*] Moreover, the reality of a perceived increase in pitch motion when fins were active was to be investigated.

A secondary objective of the fin stabilizer evaluation was the identification of possible fin system performance improvement alternatives.

The possibility of improving the fin performance by increasing fin size without altering the size of the installed machinery and by making a minor adjustment in the fin control algorithm is one particular performance enhancement alternative.

This feasible, inexpensive improvement was suggested by research in roll stabilization by means of ships rudders. Specifically, such rudder roll stabilization, RRS, systems have their performance limited by the available rate at which the rudder system is able to move in response to stabilizing commands. It has been noted during simulation work[2] that practical upper limits of rudder rates which result in roll stabilization performance levels comparable to fin systems are on the order of one-half to one-third of the maximum fin movement rates as currently installed on the BEAR class. Accordingly, it was considered important to examine the possibility of increasing the effectiveness of an installed fin system inexpensively by upgrading the size of the fins without altering the size of the installed machinery and accepting the reduced maximum fin rate.

APPROACH

FIN SYSTEM PARTICULARS

The BEAR class fin stabilizer system[3] was manufactured by Sperry Marine and installed at Tacoma Boat during construction of the first four ships of the class.

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*A complete listing of references is given on page 33.

The fin system is non-retractive and consists of a pair of fins, the hydraulic actuation machinery to move the fins, and a system of electrical/electronic controls operated from the ship's Engine Control Center (ECC). A pilothouse indicator provides the bridge with fin status and fin angle position meters.

The system's main electrical/electronic control, located in ECC, is effected by the operator at the Master Control Panel. This, in turn, is tied to the Analog Processor Unit which provides the system commands to the Local Control Units, LCU, located in both port and starboard fin spaces. It is to be noted that the Analog Processor Unit receives both the operator's commands as well as ship speed, fin position, ship roll angle, roll rate, etc. The roll angle and roll rate sensors are provided to the system by a Sperry system component known as the Auxiliary Sensor Unit, ASU.

This fin system is mechanically and hydraulically the same as the one designed and installed at this same shipyard in the same time frame on the four Royal Saudi Naval Forces, RSNF, PCG 612 class ships. The fin machinery response to commanded fin angles should therefore be essentially the same for both ship classes. However, the fin size/aspect ratio and fin controller for these two ship classes are different.

DIFFERENCES BETWEEN PCG AND BEAR FIN SYSTEMS

The 1768-ton, 255-ft (length between perpendiculars) BEAR class cutter with its 2.5 to 3.1-ft GM employs a pair of 25 square foot fins, whereas the 902-ton, 230-ft PCG 612 class ship with its 5.5-ft GM employs a pair of 30 square foot fins. These NACA 0015 section shape fins also differ in their aspect ratios, with the BEAR class employing the standard U.S. Navy practice aspect ratio of 1.00, and the PCG 612 employing the hydrodynamically more efficient aspect ratio of 1.52.

The BEAR class fin system uses Sperry Marine's standard commercial analog fin controller, brought up to military specifications. The RSNF fin system employs a state-of-the-art digital controller derived from the USN FFG 7 class fin controller. Both of these systems share the same type Auxiliary Sensor Unit. However, unlike the RSNF ships, the BEAR class system does not have back-up for the roll and roll rate sensors as part of the fin system. A fault indication signal is not provided on the BEAR system for the case when either of these sensors fail.

A back-up for the sensor signals can be derived either from the ship's roll gyro, or a second ASU unit. On the RSNF ships this back-up is provided to the fin controller from the ship's gyro by means of an operating mode switch. Again no roll and roll rate sensor fault indication signal for the crew was incorporated into this RSNF fin design.

The BEAR class fin controller, because it derives directly from the commercial fin controller, develops its fin angle command signal from a commanded fin lift signal. This signal in turn is based on roll angle and rate measured by the Auxiliary Sensor Unit (ASU) of the installed fin control system. The importance of the lift-based fin command signal is that the fin command is a direct function of the inverse of ship speed squared. Flaws in the ship speed input to the fin system accordingly have significant impact on the fin system performance.

Extensive work including sea trials were required to bring the RSNF PCG 612 fin system up to expected fin performance standards. Sea trials established deficiencies in the digital fin controller and validated the corrective modifications to the fin control algorithm developed and installed on the ship class. This work on these state-of-the art digital fin controllers has been completed, and validating sea trial results with these systems are used as the basis of comparison for the performance of these BEAR class fin stabilizers.

FIN PERFORMANCE TRIAL PROCEDURE

Fin system performance tests[4,†] involve, at a minimum, the recording of the commanded fin angle, the actual fin angle as well as the associated ship parameters including ship speed, heading and ship roll. In general, three ship speeds and three headings relative to the seas represent the minimum set of trial conditions that define the roll reduction performance of a fin system. These speeds represent a design speed of 15 to 18 knots for the system as well as one speed above and one speed below the design speed. The headings relative to the sea consist of bow seas, beam seas and quarter/following seas.

The preferred technique for evaluating the fin performance is to make the stabilized/unstabilized runs in direct succession in order to minimize variations in the underlying sea state. This is particularly true when the results are to be examined as a function of ship heading. Thus trial patterns such as the octagon

†Baitis, A.E., T.R. Applebee, and W.G. Meyers, "Seakeeping and Fin Stabilizer Performance Sea Trials with the Royal Saudi Naval Forces PGG 511 and PCG 612 Classes," Report DTNSRDC/SPD-1028-02. [Distribution limited to U.S. Government agencies and the Government of Saudi Arabia; Proprietary Information; July 1983.]

pattern[5] are much preferred over trial patterns such as the pattern employed on the second sea trial.[1] The beam sea, design speed condition represents the most important condition because it tends to produce the highest roll reduction performance for a system. The BEAR trials met or exceeded these minimum conditions and fin response measurements, and thus provide an experimental data base from which performance conclusions may be drawn with confidence.

For purposes of the analysis of the fin performance data the two sea trials tests are regarded as a series of five distinct test sequences. The first four of these sequences were performed during the first sea trial and the last sequence was performed during the second sea trial. Effectively the first two sequences represent the first two day. of testing when the first and second octagon test patterns were performed. The third sequence was conducted in order to examine the effect of a reduced maximum fin rate on roll reduction. The fourth sequence was conducted in order to establish the effect of limiting the fin angle (thus reducing fin cavitation) on roll reduction. The fifth test sequence represents the entire second sea trial.

ANALYSIS PROCEDURE

There were no expectations at the time of the trial planning and set up that the BEAR fin system might have other than a few minor, correctable deficiencies. These expectations were the result of the knowledge that the BEAR class fin system controller represented a militarized version of Sperry Marine's commercial, analog fin controller, and because of experience with a previous calm water, forced roll trial with these fins.

The initial analysis of the fin performance data following completion of the trial suggested that for some conditions the stabilization achieved was not large whereas for many of the cases the effectiveness of the fins was at expected levels. Thus it appeared that the fin system did not perform in a catastrophically deficient manner as had been measured during the RSNF PCG trials with a more advanced digital controller. Further, since experienced trial personnel from the other trials reported that beneficial effects of the stabilizer activity were clearly perceivable, no serious deficiencies with the fin performance were expected and none reported to USCG headquarters prior to the start of the second sea trial.

A more detailed analysis of the fin performance data following the second sea trial indicated that the performance of the fin stabilizer was quite poor in all but relatively mild beam seas during the first trial and that this pattern also prevailed during the second sea trial. A further investigation into the reasons for the relatively poor performance of the fin system at headings other than beam seas was therefore initiated in order to explain these performance deficiencies.

As a part of this more detailed analysis of the fin performance, the condition/operation procedure of the fin system was investigated. The condition of the system as a function of time was established by interviewing both the Sperry engineers and the DTNSRDC project staff involved. Fin status was compiled from their notes and DTNSRDC trial staff member recollections of system status at the time of both trials. Thus an attempt was made to separate various mechanical, electronic, and operational factors that impacted the measured fin performance at the various times during the first trial and second sea trial.

The procedure employed to analyze the performance of the BEAR fin system was complicated by the fact that, as this procedure was followed, the operational status of the fin system as tested was found to have been defective in various details. The six discreet stages in the analysis are listed in order of their occurrence as follows:

a. Comparison of the magnitudes of the RMS roll motions measured during the five separate test sequences of the two trials.

b. Fin angle motion algorithm development based on the fourth test sequence of tests during the first trial.

c. Fin machinery performance in following the fin command.

d. Identification of the fin command component mixture.

e. Comparison of BEAR fin machinery performance with other ships and with reduced maximum fin rate and angle.

f. Comparison of BEAR fin roll reduction performance with that of other ships.

FIN STABILIZER TRIAL SET UP PROCESS

Sperry Marine field engineers were tasked to bring fin system response signals out of the fin control system in analog form and provide signal lines for connection to the DTNSRDC recording/fin control system.

As per common DTNSRDC practice, the fin system was "groomed" or tuned to the system specifications at the outset of the first sequence of trials by Sperry Marine field engineers using standard Sperry procedures. The quality of the system tuning, however, came into some question once the first trial pattern had been completed. Generally, such system tuning involves only minor adjustments to the fin system and is accomplished rapidly and with little difficulty.

In the case of the BEAR, this system tuneup was neither rapid nor was it accomplished without difficulty. Ship's crew indicated to the Sperry engineers that the port fin would intermittently move to as much as twice the excursion as the starboard fin. The intermittent nature of this deficiency made it very difficult to track down. As a result, much time was spent looking for a probable cause of the reported fault. Unfortunately no faults were found until the ship had actually completed the first two days of sea trials.

In addition to the intermittent port fin motion, a second flaw reported by the ship was the inoperative Automatic Ship Speed Log input. It is considered likely that the improper calibration of the manual speed input signal may have resulted when someone either in the crew or in the shipyard tried various adjustments to get the port fin motion to be the same as the starboard fin motion.

The status of the speed log input to the fin system required the crew to manually set the correct speed input. Since the commanded fin angle is a function of ship speed, this deficiency in the automatic speed input to the fin system did affect the fin performance. The manual speed input was set at 15 knots and unchanged except for two runs until the speed log repair was accomplished.

It was established by the last three runs of the second octagon when ship speed was reduced to 12 knots, that the manual speed input was not properly calibrated. This improper speed input calibration thus represented the third flaw in the fin system. Evidence of this flaw or deficiency is that the maximum fin angle did not remain at the same value as for the higher 15-knot speed setting used throughout the rest of the octagon runs. The importance of the speed input to the fin system command signal is even greater here than in the RSNF ships, since obviously the fin lift command signal is multiplied directly by the inverse of speed squared.

No changes in the fin system were made until the completion of the second day's testing once the ship was diverted for operational reasons.

ANALYSIS AND RESULTS

ROLL DAMPING ANALYSIS

A ship's damping moment per unit of roll rate is an excellent indictor of the ship's propensity to roll. For a given ship of fixed metacentric height, moment of inertia and operational condition, the larger these values of damping moment are, the less the ship will tend to roll. The term "operational conditions" implies operating on a particular course, speed and sea condition. The addition of bilge keels[6] amounts to simply increasing the ship's damping moment. Similarly, it is to be recognized that the dominant term in the fin stabilization moment also adds directly to this damping moment.

Measurements of the roll damping values of the ship as tested both during calm water, forced roll trials in May of 1983, and during the March 1984 trial are summarized and presented in Figure 1. The damping data for both the 210-ft cutter CONFIDENCE of the WMEC-615 class as well as a 378-ft cutter of the WHEC-715 HAMILTON class are also provided as bases of comparison. The location of bilge keels, rudders and fins for these three principal cutter classes is shown in simplified profile views in Figure 2. Attention is focused first on the BEAR's damping results both from full-scale trials measurements and the initial design model-scale experiments. Only the data for the BEAR's 15-knot speed case is presented from the March 1984 trials.

This roll damping data was developed using the procedure of Reference 7 where comparable model scale data was presented for the BEAR class ship model without fins and bilge keels during the design cycle for the vessel. Roll damping is presented in Figure 1 as nondimensional roll decay coefficient, n, values (i.e., roll damping moment nondimensionalzed by the product of the natural roll frequency and the mass moment of inertia.) This measure of roll damping, n, is presented as a function of the average single amplitude roll angle, designated as the mean roll angle.

The predicted roll damping data of Figure 1 was calculated using the U.S. Navy Standard Ship Motion computer program designated as SMP-81.[8] This program was undergoing extensive revisions and enhancements while the work with the Bear seakeeping and fin stabilizer performance assessment trials was underway. Following the completion of a draft of the current report in August of 1984, the differences between the predicted and measured full-scale roll damping for the BEAR led to the discovery of a serious error in the coding of the roll damping subroutine. The error was associated with the damping calculated for the bilge keels when the ship section contained both a skeg and bilge keels. The correction of this error as well as other improvements in the roll damping theory and the inclusion of rudder and roll stabilizer prediction capability were subsequently completed and reported in Reference 9.

Following the completion of this latter work, the revised Standard Ship Motion computer program designated as SMP-84 was then applied to the BEAR roll damping data. SMP-81 roll damping results are shown as solid lines in Figure 1, whereas the SMP-84 results which supersede these older, incorrect results are shown as dashed lines. The damping data shown in the tables only reflects the corrected SMP-84 calculations.

BEAR Roll Damping

The comparison between the measured full-scale roll damping and that predicted by SMP-84 illustrates good agreement and highlights the magnitude of the error in the predicted damping using SMP-81. Although other experimental roll damping from model tests[7,10] and full scale trials (see page 4 footnote) exhibit a stronger speed dependence of the roll damping than is evident from the BEAR's full-scale measurements and predictions, these data do not appear to be unusual in any way. However, a comparison between these BEAR trials damping data with model-scale roll damping of Reference 10 in similar load conditions at 15 knots indicates substantial differences in magnitudes between the model-scale and the full-scale roll damping data.

Since these model tests were made at the time of the design of the ship and were used then directly in the sizing of the bilge keels and fins, the differences between the prebuilt damping predictions (with model tests) and the final as built ship damping are significant. The measured model-scale roll damping for the ship without bilge keels and fins is equal to or slightly greater than that for the actual ship with bilge keels and fins. The BEAR's actual roll damping is therefore lower than would be expected from the model-scale measurements. The model data overstates the roll damping of the ship and thereby underestimates the required roll damping to be provided by the fin stabilizers at moderate to high speeds and by the bilge keels at low speeds.

As a result of these differences between model and full-scale roll damping, it is concluded that the design sizing procedures of Reference 10 for the fins and bilge keels of the BEAR class has resulted in fins and bilge keels that are too small.

Components of BEAR Roll Damping

The importance of the difference between model-scale and full-scale roll damping coefficient, n, may be inferred from the calculated magnitude of the damping developed by the various appendages and the hull. Table 1 provides a breakdown of the damping components for the BEAR as well as two other primary cutters, i.e., the WHEC-715 and the WMEC-615. The length and location of the bilge keels, rudder and fins are similarly shown in the profile of these ships in Figure 2 with the additional specifics of these roll-reducing appendages being given in Table 1.

A review of the BEAR's SMP-84 damping data indicates that the predicted damping consists of the damping due to four major terms: damping due to the hull and skeg (0.041), damping due to the rudder (0.032), damping due to bilge keels (0.025) and damping due to the inactive fin (0.031). The difference between measured model- and full-scale roll damping represents almost one half of the total damping. In effect, the fully-appended, full-scale ship exhibits the damping characteristics of the model without either bilge keels or fins.

The question thus arose as to why such large differences occur. An examination of the natural roll periods associated with the measured full-scale and predicted (SMP-84) roll damping was therefore initiated to ascertain if measured and predicted roll damping really was for the ship in the same load condition and, furthermore, to define the sensitivity of roll damping and natural roll period to variation in ship load conditions represented by the experimental data.

It is to be noted that the data in Figure 1, except for the 15-knot case, represents the BEAR on its last leg of the trip from Tacoma Boatbuilding Company, Tacoma, Washington to the Coast Guard Yard, Curtis Bay, Maryland in 1983. The 15-knot data on the other hand represents the BEAR in March of 1984 after leaving Curtis Bay, having been reballasted and otherwise completed for service. In fact, this 15-knot data is further subdivided into tests made on 5 March 1984 (just before the rough water sea trials) and tests made on 13 March 1984 (at the conclusion of the sea trials shortly before returning to Portsmouth, Virginia).

The measured roll damping and corresponding natural roll period data for each of the test conditions is shown in Figure 1. The roll period data was rounded to the nearest tenth of a second. For the 15-knot case, roll damping corresponding to the pre-trial roll period is designated by an open triangle and roll damping corresponding to the post-trial roll period is designated by an open square. In this context, it is to be noted that variations on the order of one-half second are within the scatter of the results from a series of tests at the same ship speed within a given day. It is evident that the ship's natural roll period appeared to increase by about one half a second during the period of the trial as a result of the variations in the ship's loading.

The roll period for the BEAR during the 1983 trial ranged from 11.1 to 11.6 seconds as ship speed varied from 12 to 17 knots. It is noted that this slight increase in the roll period with the increase in ship speed is as expected, whereas the 0.3 second drop in the period from 11.6 seconds at 16 knots to 11.3 seconds at 17 knots is considered to be representative of the experimental resolution.

The lack of a substantial difference in the roll period data for the ship in 1983 in its pre-ballasted condition and the March 1984 trials suggests that the ballast change did not affect KG [GM] or alternatively that the alteration in KG was offset by a change in the roll gyradius of the ship. The load condition for the ship as defined by the March 1984 trials was accordingly used also for the damping predictions of the ship as tested in 1983. It is evident that the SMP-81 damping predictions for the ship do not agree with the measurements and that these differences must be attributed to deficiencies in the basic theory.[7,8,9,11] The corrections in these deficiencies are similarly evident.

HAMILTON, BEAR and RELIANCE Class Roll Damping Comparison

The best available, full-scale trial roll damping data for the 210-ft WMEC was accordingly examined and contrasted with predicted results. It is noted that these data were taken from trial results with the WMEC 619 in Chiniak Bay, Kodiak, Alaska in 1982. A comparison of the rather limited measured roll damping with SMP-81 damping predictions in Figure 1 suggest that damping results for this ship class are also underpredicted. Again, the repeated calculations of roll damping for this ship with the SMP-84 program bring the sparse experimental data and this revised theory into apparent agreement.

Roll damping predictions for the 378-ft cutter were also made in order to place the damping for the BEAR into context with the other members of the major cutter classes. Measured roll damping data unfortunately was not available for the 378-ft cutter. It is evident from the data of Figure 1 that the roll damping of this 378-ft cutter is quite similar to the roll damping of the 210-ft cutter, and furthermore that the BEAR's damping is low relative to both of the older cutters.

Table 1 was therefore prepared to assist in determining the probable reason for this fact. The total roll damping as predicted by SMP-84 is presented in terms of the four major components, along with the related ship particulars to make it possible to dimensionalize the damping. Finally, the particulars of the appendages which appear to be the reason for the rather low roll damping of the BEAR are also presented. All damping data in this table is for a 15-knot ship speed in order to facilitate a direct comparison between the ships.

One glaring difference between the roll damping values for all three of these cutters is very apparent, and that is the very large differences in the roll damping provided by the bilge keels of these vessels. Both the HAMILTON and the small RELIANCE class cutters exhibit comparable roll damping characteristics for the bilge keels. Also for both of these ships, the bilge keels provide the dominant component of the total roll damping.

Clearly, the BEAR is very different from these two vessels. The BEAR with its proportionally much smaller bilge keels and the less than optimal[6] location of the bilge keel relative to the fins derives only a relatively small portion of its total damping from these bilge keels. In fact, even when the damping due to the passive fin is added to the bilge keel damping, this total is still only one-half the damping due to bilge keels alone for either one of the older cutters. It is concluded, therefore, that the BEAR with its small bilge keels has its low speed and/or inactive fin roll performance penalized as a direct result of this fact. In this connection it is also to be noted that the performance of the active fins in turn is penalized by the location of the bilge keels aft of the fins, as noted in Reference 6. Such placement of fins degrade the lift generation capacity of the fins by very substantial amounts and thus effectively "reduces" the fin area and roll stabilization potential.

The size of the bilge keels on the BEAR could be increased with potentially good payoff in a reduced roll motion particularly at lower ship speeds where the fins are ineffective.

BEAR BILGE KEEL AND FIN SIZE INCREASES

A brief series of runs was made with the SMP-84 program to quantify the potential payoff of a bilge keel size increase, a fin size increase, and the combination of both in terms of the increase in the passive roll damping and the consequent roll. The resultant roll is also presented for both active and inactive fins. These initial efforts were made to identify options for altering the BEAR's current roll damping characteristics to fall more nearly in line with proven practice, and employed only the most feasible, inexpensive and obvious choices.

It is to be recognized that the benefits of the fin size and bilge keel area increases accrue primarily at the higher speeds for fins and at the lower speeds for bilge keels. Calculations were, however, made only at a single moderate speed of 15 knots.

Fin area increases and bilge keel span increases are assessed in terms of roll damping for a 5-degree mean roll angle and are presented in Table 2. The corresponding RMS roll motions of the BEAR are also given in the table for active and inactive fins. The BEAR was assumed to be traveling at 15 knots in a 13-foot significant wave height, 9-second modal period, shortcrested beam sea.

A bilge keel span increase of 1.0 ft (from 2 to 3 feet) is considered to be a feasible, relatively inexpensive way to achieve a bilge keel increase without altering the location of the bilge keels relative to the fins. Increasing the fin area to 40 square feet is similarly considered to be the most feasible and inexpensive fin growth possibility. This increase in fin size can be achieved by scaling the current fin up to the geometric limitations imposed by the design requirement not to extend the fin below the ships baseline nor outside of the 5-degree static heel at the pier. This simple fin size alternative was initially investigated as part of the fin sizing design effort.[10]

From the predicted roll damping results of Tables 1 and 2, it is evident that the changes in the bilge keel and fins do not bring the BEAR's total roll damping into line with that of the older cutters. The low speed roll damping of the BEAR therefore still needs to be increased.

In fact, the impact of the added bilge keel span amounts to only about 6 percent reduction of roll with the fins inactive. When the fin size is increased to 40 square feet and fins are inactive, the RMS roll of the ship is reduced by another 6 percent for a total of 12 percent from the current ship. Of course, at lower ship speeds these benefits would be larger.

Although these simple, feasible increases in the passive damping of the BEAR at 15 knots do not result in what may be judged to be a very satisfactory improvement, it should not be concluded that this approach must be abandoned. The hull passive roll damping should be increased and the issue is how to achieve this relatively inexpensively. Alternative measures need to be developed.

As an indication of two such an alternatives, consider, for example, the employment of the largest possible fin which the present machinery including bearings, tiller, etc. could support. Such additional fin growth can be achieved by increasing the chord once the baseline and 5-degree heel lines are reached with the 6.325-ft span of the 40 square foot fin. For details refer to Appendix A. In other words, make the fin as large as possible constrained only by the machinery limitation. Secondly, consider using the "old" 25 square foot fin as passive anti-roll surface mounted forward of the fin location, or alternatively consider adding a limited amount of bilge keel forward of the existing fin. It is considered that the careful implementation of these latter alternatives, or variations thereof, can be employed to bring the passive roll damping of the BEAR class hull into line with that of the older cutters. That is, bring this ship class' low speed roll motion characteristics into agreement with these other cutters.

In the context of improving the BEAR's passive roll damping by the addition of either the "old" 25-ft fins or the largest possible bilge keels forward of the fins consistent with not introducing pitch/slamming effects, it is to be recognized that such additions will not detract from the performance of the fins. Location of the fin forward of the bilge keels unfortunately results in very definite degradations in the active lift generated by the fins as bilge keel size is increased. For the 2-foot span bilge keel on the BEAR, the fins lose about 16 percent of their lift-generation capacity at 15 knots. As the bilge keel span is increased to 3 feet, the lift generation is decreased by 33 percent compared to when there is no bilge keel aft of the fin.

When the stabilized performance of the BEAR is examined with these feasible alternatives of the 3-foot span bilge keel and the 40 square foot fin, the predicted improvements due largely to the fin area increase are clearly illustrated by the stabilized roll data of Table 2. The stabilized roll of the ship in its "As Is" condition is reduced by an additional 40 percent from its current stabilized value of 2.9 degrees. The total roll reduction thus achieved by the increased bilge keels and fins is 65 percent, to a negligible 1.7 degrees from a rather severe 4.8 degrees. This level of active roll stabilization provided by an upsized fin system for the BEAR class should yield a ship ride with as small roll motions as the stabilized U.S. Navy FFG-7 class, at a fraction of the investment.

It is to be noted that an increase in fin size will also improve somewhat the active stabilization capability of these fins at the lower ship speeds where the present fins are ineffectual. An increase in fin size is likely to be less expensive than the purchase of a power unit and controller to provide roll stabilization by means of the rudders.

INFLUENCE OF FIN ACTIVITY ON VERTICAL SHIP RESPONSES

One area of concern communicated to DTNSRDC prior to the start of the sea trials was the reported influence of the fin stabilizer activity on the pitch and vertical acceleration responses of the BEAR. It was considered that this influence would have to be very small since the size of the fins and associated fin forces and moments are very small in relation to the wave disturbing forces and moments that produce the vertical plane response of this ship.

Figure 3 was prepared in order to address the impact of fin activity on the vertical responses of the ship. Specifically, a summary of the pitch responses is presented for three major test sequences of the trials. Two measures of the pitch responses are presented. The statistically stable RMS pitch response is shown as the shaded area in the bar graphs of the figure and the less stable maximum responses within an individual test run are shown as the unshaded portion of the bar graphs.

A comparison of the pitch responses for pairs of runs with active and inactive fins can be made to establish the influence of fin motion on the pitch of the ship. However, such a comparison requires consideration of the expected results in order that a misinterpretation of the very limited data not occur.

What should be expected is that in one-half of the cases the pitch will be slightly greater when the fins are activated than when they are inactive. Thus, if fin activity has no impact on the pitch motion of the ship, in half of the cases, the unstabilized ship will be larger in pitch than for the stabilized ship. Furthermore, although a larger RMS pitch motion should, in general, yield a larger extreme pitch motion, the statistical sample variability of these extreme motions from one run to the next may mask the impact on pitch motion due to the fin activity. Similarly, if the fin activity effect is small, the inevitable variation in the sea state from one run to the next may also overshadow this effect.

In each of the six pairs of runs (fins on/fins off) for the first trial sequence on 6 March, the maximum pitch during the stabilized run exceeds the value for the unstabilized ship. The associated RMS pitch responses in three out of the six pairs of runs are larger for the stabilized than the unstabilized case. Two of the remaining pairs of runs have the same RMS pitch values for the stabilized and unstabilized cases and the last pair of runs have smaller RMS pitch for the stabilized than for the unstabilized case.

This set of results in Figure 3 therefore suggests that there is some additional extreme pitch motion for the active fin case. It is not clear, however, whether or not this is due entirely to the activation of the fins or whether the fact that the seas were continually decaying during the test period influences the results.

A similar examination of the last two test sequences (7 March and 10 April) of Figure 3 does not indicate that the activation of the fins affects the pitch motions of the ship. In one case, for three out of the eight pairs of runs, the maximum pitch is greater for active rather than inactive fins, and in the last case, for exactly half of the cases, the maximum pitch is greater.

A power spectrum analysis of several pairs of runs was made to investigate the possible correlation between ship roll and responses such as vertical acceleration, pitch and yaw. No such correlation that would be indicative of fin activity influencing vertical plane responses was noted. No possible mechanism for the fin motion coupling through roll motion into the remaining ship motions was therefore identified.

The data thus summarized does not indicate that the activation of the fins affects the vertical plane ship responses. There may indeed be cases such as the one presented for the first test sequence where the activation of the fins is associated with slight increases in the pitch motion of the ship, but it is not clear either that the fins caused this increase or that it is of perceptible significance. It is considered more likely that the apparent increase in pitch is merely the perception of an increase in pitch once the roll motion has been reduced.

RMS ROLL REDUCTION PERFORMANCE

The performance of the fin system in reducing ship roll and the associated improvement of the habitability of the ship (i.e., the reduction in the transverse acceleration levels) may be judged on the basis of two statistical measures of ship responses: RMS ship responses and maximum values of these responses. The incidence of serious motion-induced accidents is directly related to the maximum values of the responses, whereas the general long-term fatigue and work rate degradation effects[2,5,12] may best be judged on the basis of RMS responses. It is to be noted that these latter effects were not measured or directly investigated because such an evaluation requires trials that are of long duration and not suited for the short fins on/off work as performed on the BEAR seakeeping and crew/fin performance trials.

Unfortunately, though transverse accelerations at various crew work stations were measured, only the roll motions have been employed in the analysis of the fin system performance. The operational status of the fins sufficiently complicated the analysis to make it necessary to delete a detailed examination of the transverse acceleration data.

Figure 4 presents a summary of three extensive trial sequences in terms of RMS roll versus ship heading relative to the waves. The term RMS roll refers to the Root-Mean-Square motion level which is a stable statistical representation of the motion level encountered. The data are given in pairs of runs with the unshaded bar representing the fins on condition and the shaded bar represents the fins off condition. The first two trial sequences refer to the March trial and the third sequence refers to the second trial in April of 1984.

These results clearly indicate, particularly in quartering seas, that the roll reduction achieved by these fins is very low at times. In fact, during the first trial sequence the fin system actually destabilized the ship in roll. The lack of consistent, expected roll reduction performance trends indicate that the system is definitely malfunctioning. An analysis of the controller command signal and fin motion was therefore undertaken in order to identify the operational status of the fins and the reasons for the malfunctioning of the system.

FIN SYSTEM ANALYSIS

Roll Control Algorithm

The proper phasing of the fin motion so as to reduce the ship roll is achieved by developing a fin position command which opposes the roll excitation moment produced by the action of the waves on the ships hull. In general such a fin control algorithm is developed directly from fin lift measurements or ship roll measurements. In the case where the control algorithm is based on roll motion, the algorithm generally consists of a mixture of roll angle, roll rate, and roll acceleration.

It is to be noted that in beam seas the magnitudes of the roll acceleration and the roll angle terms should largely cancel since under these circumstances it is the roll rate which represents the best roll reduction control algorithm. In other words, in beam seas the roll rate is the best estimator for the wave-induced roll moment that is to be reduced.

In quartering seas, the above rule of thumb is no longer valid because roll motion is no longer produced predominately by the wave-induced roll moment. In these longer period seas, the wave-induced yaw rate moment feeds strongly into the roll motion. This longer period roll excitation is the second major component in the roll production that is also to be cancelled by the action of the fins or rudders in an RRS system. The roll reduction control algorithm in quartering seas, therefore, must cancel both the wave-induced roll moment and the wave-to-yaw rate-induced roll moment. This latter objective can be obtained by increasing the relative amount of the roll angle component of the stabilizing command signal.

In bow seas, the relative importance of the roll acceleration component increases as the roll motion period becomes shorter due to the shorter encountered wave periods. It is not clear at this stage whether the importance of the roll acceleration term is now increased because the wave moment producing the roll motion leads the wave-induced roll moment or its estimator, roll rate, more than at roll resonance; or whether the acceleration term lead is helpful in permitting the fin machinery to better cope with the higher fin rate commanded.

It is, nevertheless, an observed fact that the deletion of the roll acceleration term in bow seas does not particularly alter the performance of an RRS or fin system, whereas the deletion of the roll angle term will significantly degrade the performance of both RRS and fin systems in quartering seas. The latter observation rest on both the PCG/PGG fin trial results (see page 4 footnote) and the BEAR/HAMILTON fin/RRS trial results.

Roll Controller Command Signal Component Analysis

An analysis of the fin system performance was initiated based on the fact that the control algorithm employed on the BEAR was known to consist of a mixture of the three signal components, all of which were based on the roll motion of the vessel. The roll angle, roll rate, and roll acceleration terms thus form the basis of the roll control signal. The gains associated with these terms are referred to as K1 for the roll gain, K2 for the roll rate gain, and K3 for the roll acceleration gain.

A comparative analysis procedure was used to establish the composition of the fin command signal. This procedure employed the results from both the time domain and the frequency domain. The time domain analysis utilized the distribution of amplitudes as defined by the individual cycles of the time history (identified by three successive zero crossings); the frequency domain analysis, on the other hand, utilized the distribution of the energy of the frequency components of the time history as defined by the calculated spectral ordinates. The process is a diagnostic tool for examining the performance of a fin control system and is not a controller design tool.

Specifically, both the individual cycles of the time histories and spectral ordinate data were used in an iterative process to establish the mixture of the control signal components. As a first order estimate of the signal mixture, the unfiltered and unclipped data were mixed in the frequency domain in order to define relative magnitudes of the various components. Next the first resulting K1, K2, and K3 values were refined by going to the time domain results and applying clips and filters on the derived command signal until the results nearly agreed with the measured command time history.

The total number of cycles in the derived and calculated fin command time histories as well as the distribution of the response cycles, including the maximum values, were used to develop the effect of clipping and filtering on these signals. Once the time domain conditions were met, the correlation between the measured and calculated fin command signal became very high. This high correlation between signals was evidenced by comparing the time histories of the two signals.

The spectral shape of the measured and derived command signals were compared for agreement in spectral amplitude in order to approximate the ratio of the mixture of control signal components. Once a reasonable agreement was attained, the phases between the measured and computed signal spectra were examined. In principle, the phase angle between the two signals should be very small, with this phase representing the minor differences in the specific filter characteristics used on the components of the measured fin command and on the sum of the calculated fin command. Adjustments in the phasing were then made by altering the relative amount of the acceleration component of the mixture.

As a final check, the coherency between the two signals was examined and found to be very high. Since both control signals were derived from the same physical signal, this high coherency provided the evidence that the derived fin control signal composition had been correctly established. The differences remaining between the calculated and measured command signal then may be attributed to differences in the actual sensors used by the measured command and the specific filters employed on the fin components in the controller.

The application of these techniques to the fourth test sequence of the first trial indicated that

a. the fin control law or algorithm used appeared to employ only roll rate, and

b. the control law did not vary within the test sequence.

The subsequent application of this control law to all of the other trial sequences from both trials then indicated that the same control law and thus the same gain also applied to the remaining data.

On the basis of these results and a subsequent meeting with the Sperry Marine design engineering staff, it was concluded that

a. the fin system had been operated throughout both sea trials in the MANUAL GAIN mode,

b. the proper operation of the fin system is made sufficiently complex by the types of switches used that not even a design engineer from Sperry, nor a service engineer from Sperry, noted the improper MANUAL GAIN setting rather than the correct AUTOMATIC GAIN mode operation of the fins,

c. a simple fin operator guidance sign or placard should be affixed to the control console to prevent the improper operation of the fins by the operator.

Fin Machinery Performance in Following Command Signal

Following the initial fin command component analysis, the entire process was repeated and fin machinery response was contrasted with the measured fin command. The results were used to further refine the derived signal mixture and to establish the fact that the machinery was operating properly. As a consequence of the machinery performance analysis, the magnitude of the acceleration component of the signal was refined in relation to the more dominant roll rate, and the absence of the roll angle component of the control signal was confirmed.

The analysis for the machinery operation utilizes both the range of periods over which the machinery operates and the degree to which this machinery satisfactorily follows a command signal. The range of periods of operation over which fin machinery must satisfactorily follow a fin command signal may be referenced to the natural roll period of the ship. The dominant roll motions of the ship occur at roll periods equal to or greater than this natural roll period. Periods longer than the natural roll period are easier for the machinery to follow since, at these periods, the limiting machinery hydraulic flow rates and associated power limitations are less frequently encountered than at the natural roll period. At operating periods less than the natural period, the power and flow limitations are incurred more frequently than at the natural roll period.

In general, it is bow sea fin command signals which represent the limiting machinery operating conditions, whereas it is in beam or quartering seas with the longer roll periods where the maximum ship roll motions occur.

Typical results of the machinery analysis are presented in Tables 3 and 4 as well as in Figure 5. The tables document the power spectra of fin or rudder command, designated as S1, and fin or rudder angle, designated as S2 for the BEAR, the CONFIDENCE, and finally the base ship, the PCG.

The maximum, numerically-valid spectral ordinates and their associated frequency range were defined as values that were equal to 5 percent of the peak spectral ordinate. Both the peak spectral ordinates and the valid range of the ordinates are marked in the tables. The 5 percent range was considered to define the minimum range of machinery response frequencies from which valid conclusions about machinery operational performance could be deduced.

The examination of the amplitude corresponding to the 5 percent reliability range spectral ordinates indicated that these values were much greater (8 times in fact) than the basic measurement and recording resolution. It was therefore concluded that if the coherency is still very high (above say 0.7), the spectral ordinate data still contains useful information.

When the machinery phase and coherency data for these runs was further examined with this above relaxed rule (Table 3), it became apparent that for these examples of the BEAR fin motion, the data may be regarded as valid all the way down to motion periods of 2.2 seconds (run #45) and down to 2.4 seconds (run #43). Thus this data illustrates that the range of motion periods to which the fins can be driven accurately for control purposes is almost unaffected when the maximum fin rate is reduced to 20 degrees per second.

Thus while it is the phase lag that determines how well the fin follows the control signal, it is the coherency criteria (> 0.7) by which the reliability of this phase data can be judged.

On the other hand, when the same machinery as installed on the PCG class is examined (see Table 4) it becomes clear that, even where much higher fin machinery demands are placed on the system, the fins can be driven with good fidelity on this vessel down to periods of 2.7 seconds. It is to be noted again that this PCG machinery has the same 42-degree per second capability as the BEAR system.

The fin machinery on the BEAR therefore functions down to periods that are 20 percent of the natural roll period, whereas the same machinery which produces satisfactory roll reductions on the PCG operates down to periods that are 38 percent of the natural roll period. It is apparent that the fin machinery installed on the BEAR class is less stressed than as installed on the PCG class.

Machinery Response Lags: BEAR Sea Trial

The fin angle response (see Table 3) with the normal, maximum available fin rate lags the fin command by 10.4 degrees or less for fin response periods ranging all the way from 42.7 to 6.4 seconds. Further, at the roll resonance period of 11 to 11.5 seconds, the fin motion lags the command only by about 6 degrees and exhibits a coherency of one. In fact, excellent coherency between the fin command and the actual achieved fin angle is exhibited over the entire range of responses. This indicates that the fin angle is faithfully following the command signal.

The fin angle response with reduced maximum fin rate lags the fin command by a much greater 27.4 degree or less than when the normal maximum fin rate is available. At resonance the fin with reduced rate lags the command by about 16 degrees rather than 6 degrees. A comparable loss in the coherency between fin command and fin angle is not evident. Thus despite the somewhat slower fin response to the fin command at the reduced fin rate, the fin still faithfully follows the command signal.

In order to assist in the interpretation of the importance of this characteristic fin machinery response lag, the same data is provided for the PCG. This data was obtained on the April 1984 sea trials, after this system was repaired and its digital controller modified. It is to be noted that the natural roll period of this vessel was much shorter at 7.0 seconds. As a result, the PCG'a fin machinery, although identical to that of the BEAR, was commanded to move the fins at higher frequencies.

Figure 5 presents some of the tabulated results from Tables 3 and 4 in graphical format. A graph of the magnitude of the phase lag with which the fin motion follows the fin command as well as the coherency between fin motion and the fin command is presented. This data is shown as a function of the period of the signals.

It is apparent that the fin machinery when installed on the PCG operates over a much narrower range of periods than when installed on the BEAR. When this machinery is operating properly as indicated by the satisfactory roll reduction performance of the fin system on the PCG class, the machinery phase lag lies between the lag for the normal rate of the BEAR fin system and the reduced rate. Further, it is noted that the coherency of the fin motion on the PCG is as narrow as the phase lag graph. This range of validity of phase lag was determined from the range of validity of the measured PCG fin responses, i.e., the range of valid spectral ordinates. It is apparent, therefore, from this machinery performance data (magnitude of lag and value of coherency over range of fin motion) that the BEAR's fin machinery was operating properly at the time of the trials.

Figure 6 was prepared to demonstrate the machinery lag in the form of a time history of the fin command, the port fin angle response and the roll motion of the ship. This data illustrates that the fin command did require the fins to move out to the maximum angles and thus is representative of fin motion levels that are commanded under more severe ship motion conditions.

The short segments of the time history are shown in Figure 6 for the fin with the normal and reduced maximum rates. These histories are particularly instructive for the first 10 seconds where the fin is ordered through a nearly complete maximum to minimum fin angle travel of ± 20 degrees.

Attention is drawn to the first full upward movement of the fin command from minus to plus 24.8 degrees. In order to track this upward movement of the fin command, the main hydraulic fin pump is ordered by the fin command to go to full pump stroke and to deliver the maximum pump output flow. In the case of the reduced fin rate, this pump stroke command has been effectively halved. This reduction in the pump stroke command was achieved by the Sperry design engineer inserting a resistor into the pump stroke command circuit to limit the stroke command.

It is evident in the top graph of Figure 6 that the port fin has little difficulty in following the fin commands as developed from the ship's roll when the maximum normal fin rate of 42 degrees per second is available. The fin accurately follows the command as is illustrated by the fact that the fin position lags the fin command by no more than 0.61 seconds in reaching the limiting fin angle. As the maximum fin rate was reduced (see the bottom graph of Figure 6), the fin lags the limiting fin angle command by about 1 second. The importance of this extra lag in the fin motion therefore needs to be determined.

Importance of Fin Lag Due to Reduced Maximum Fin Rate

The importance of the lag due to the reduced maximum fin rate may be inferred from a contrast between the fin lag data of the BEAR with normal and reduced fin rates and comparable data from the PCG. This data was obtained from the same machinery that produced satisfactory roll reduction performance on the PCG.

In Figure 5 the natural roll periods of both ship classes are clearly indicated. It may be noted that the phase lag of the PCG is both greater than that of the BEAR and more erratic. The greater phase lag of the PCG is the result of a filter installed in the local control unit of the PCG. This filter was installed in order to "protect" the pump from extraneous fin command signals due to the internal electromagnetic interference (EMI) of this ship. The need for this type of filtering was not apparent for the BEAR. The erratic nature of the phase lag of the PCG as compared to the BEAR is the result of having filtered the BEAR data during the data analysis. The BEAR's phase lag data was thus smoothed during the analysis whereas the PCG lag data was derived from unsmoothed measurements.

If the PCG phase lag data were to be shifted so that the natural roll period of the PCG were to coincide with that of the BEAR, then the relative importance of the phase lag due to the reduced fin rate can be inferred from the data. At the critical motion periods that are shorter than the natural roll period, the 42 degree per second PCG fin system lag is somewhat greater than that of the 20 degree per second BEAR fin system. Since the PCG fin system performed satisfactorily in reducing ship roll, it is therefore inferred that the BEAR's fin machinery system will also produce satisfactory roll reductions when the maximum fin rate is reduced to 20 degrees per second.

The machinery performance of the 8 degree per second steering gear of the 210-ft WMEC CONFIDENCE as shown in Table 4 represents a further point of reference by which the BEAR fin machinery adequacy can be judged, particularly at reduced fin rates associated with a larger fin. It is noted that even here, where the machinery was able to adequately track the command signal only down to around 8 seconds, or 70 percent of the natural roll period, this proved to be adequate for roll stabilization purposes. The levels of roll stabilization on the order of 25 percent achieved with this RRS system were limited only by the control algorithm and the hydrodynamic coupling between roll and yaw, and not machinery characteristics. It may be concluded that the accurate, timely (without large lags) fin system response down to ship roll periods equal to 38 percent (PCG) or 20 percent (BEAR) of the natural roll period of the ship represents a superfluous machinery response capability not required for satisfactory roll stabilization.

The conclusion drawn from the machinery data of the three ship classes presented is that it is feasible to increase fin size and reduce the maximum fin rate to 20 degrees per second without altering the required machinery (power), or the capability of such a system to satisfactorily reduce the ships roll motion. Such an increase in the roll stabilizing capacity of the BEAR class fins is therefore a viable option.

It is to be determined what impact the response lag resulting from the reduced fin rate has on the roll reduction performance of the fin. It is after all the reduction of the roll motion which is the best measure of the importance of available fin rate in the roll stabilization of this BEAR class cutter.

Effect of Decreasing Maximum Fin Rate

The influence of a reduction in the maximum available fin rate on roll reduction was examined with a series of port and starboard beam sea runs in seas ranging from five to seven feet. That is, the influence of a reduction in the available fin rate was examined at 15 knots in available sea conditions. These mild seas did not often require very substantial fin angles.

Direct evidence of the roll reduction capacity of the BEAR's fin system at reduced maximum fin angle rates is therefore shown in the top graph of Figure 7. The data is again shown in the same format as the summary data of Figure 4. Since the trial sequence was conducted successively in both starboard and port beam seas, the results are shown side by side for direct comparison purposes.

The starboard beam sea data suggest that a small decrease in roll reduction performance is associated with the reduced maximum fin rate, whereas the port beam case suggests that a small increase in the roll reduction performance is associated with the reduced maximum fin rate. This slight scatter in the results is considered to be indicative of the experimental accuracy/repeatability. It is concluded based on these limited results that the roll reduction performance is not particularly affected by the reduction of maximum fin rate investigated.

Despite the drawbacks associated with the mild seas, the experimental machinery and roll motion results suggest that the degradation of roll reduction performance with a decrease in maximum fin rate from 42 to 20 degrees per second is so small that it does not appear to penalize the roll reduction performance of the fin system. These results agree in general with the simulation work performed as a part of the PCG/PGG program by Sperry Marine in preparation for the BEAR trials. Additional aspects of the work supporting the fin rate impact on roll reduction were reported by Sperry Marine in an informal report entitled "Aspects of Roll Stabilizer Performance on WMEC 901". Appendix A was similarly prepared as additional fin size alteration information.

In conclusion, these limited results suggest that substantial increases in the performance of the existing fin system can be achieved by increasing the size of the installed fins to limits governed by the structural strength of the fin shaft and bearing assembly. Moreover, the associated reduction in maximum fin rate is not likely to affect the roll reduction performance.

Auxiliary Sensor Unit (ASU) Failure Detection

As the fin machinery performance was examined using the phase lag between the measured fin motion and fin command, a series of such phase lag calculations were made wherein the measured fin motion was contrasted with the calculated fin command. This calculated fin command had been based on the control algorithm which contained only roll rate (see Roll Controller Command Signal Component Analysis).

The phase lag results for this comparison of measured fin angle to calculated fin command indicated that the fin angle led the calculated command by a substantial number of degrees over the entire active range of fin motions. This result essentially indicated that the fin motion occurred before the calculated fin command. Clearly this represented a paradox. No known combination of physically realizable filters in the Sperry fin controller could account for this set of results.

Accordingly, a second extensive meeting with the Sperry design engineers was held. During the course of this working meeting, the addition of a roll acceleration term into the calculated fin command was suggested, tried, and found to cancel the lead of the measured fin motion over the calculated command. It was therefore deduced from this result that the command signal on the BEAR was missing only the roll angle term. Further, this missing term accounted for the poor roll reduction performance of the fin system in quartering and following seas.

The fact that the ASU roll sensor had not been operating during the first and second trials places this failure in the same category as the operation of the fin in the MANUAL rather than AUTOMATIC GAIN mode. It is evident that the fin check-out procedure employed by either the ship or fin service personnel must include the check-out of the individual ASU sensors. Such a check-out procedure should be incorporated as part of the system inspection prior to sailing as a matter of general operating procedure.

Fin Cavitation and Maximum Commanded Fin Angles

During the first two days of sea trials, the fins were found to cavitate rather severely as evidenced by obvious and clearly audible noise. Additionally, reports from the ship that the paint had been worn off the fin edges substantiated these observations. Accordingly, the fin controller was adjusted to call for maximum fin angles of 20 degrees rather than 24 degrees. The fact that this reduction in the maximum commanded fin angle did not degrade the roll reduction performance was documented by the results from trial run numbers 57 through 59 as shown in Figure 7b.

It is to be noted that this alteration in the maximum commanded fin angle represents a BEAR class fin controller change that should be made on each of the ships of this class.

The fin system on the BEAR was accordingly brought up to this status by the end of the first sea trial. The fact that the ASU's roll angle sensor was not operational was not known at this time. Repairs/adjustments of known defects were accomplished during the times when the trial crew could not test due to a multi-ship operation which included the BEAR.

BEAR FIN PERFORMANCE COMPARISON WITH OTHER SYSTEMS

Some limited, recent roll stabilization data for the PCG and the 210-ft WMEC CONFIDENCE is presented in Table 5 along with comparable data for the BEAR. This data is presented for the same 15-knot ship speed in 6 to 8-foot seas and represents a series of roll reduction performance levels judged to vary from unsatisfactorily low to acceptable.

RMS roll motion data is presented for headings that ranged from bow to quartering seas. Bow and quartering seas referred to the cases when the seas approached the ships from 45 degrees off either the bow or stern, respectively. It is to be noted that the seas in which the PCG was tested were short, freshly wind-generated seas that contained no underlying swell, whereas the seas for both the BEAR and the CONFIDENCE were longer, open ocean seas which did contain underlying swell.

It is also to be noted that the fin system of the PCG had been modified and was operating at full efficiency at the time of the trials. The performance level of this ship's fin system was judged to be satisfactory and thus represents the basis of comparison for the BEAR fin performance.

The bow sea data for the PCG illustrates a relatively low roll reduction of 22 percent which is attributed to the low level of roll encountered (less than one degree) and to the relatively short encountered roll periods. The BEAR's 43 percent roll reduction in bow seas is indicative of an acceptable, though not overly high, roll stabilization performance. The repair of the BEAR's ASU roll sensor as well as the proper operation of the fin system in the AUTOMATIC GAIN mode is likely to increase this bow sea roll reduction performance slightly.

The beam sea data for the PCG illustrates satisfactory roll stabilization performance of 53 percent which is essentially matched by the stabilization performance of the BEAR fin system. Alterations in the repair status of the BEAR's fin system is not likely to increase the roll stabilization of the system in any significant way in beam seas. It is noted in this context that this level of roll stabilization is somewhat better than the roll stabilization achieved with the first generation RRS system installed aboard the WHEC JARVIS and MELLON. Thus this level of performance is considered to be the lower bound of acceptable roll stabilization for a fin system.

The quartering sea data for the PCG illustrates a 56 percent level of roll reduction performance that matches its performance in beam seas and corresponds to the level of performance to be anticipated for the BEAR system once the malfunctioning ASU roll sensor is repaired. The present level of roll reduction performance of the BEAR's fins in quartering seas is even lower than that of the CONFIDENCE which has been judged to be unacceptably low even for an inexpensive RRS backfit rather than a fin system. In this context, the 7 percent roll destabilization obtained with the BEAR system at 19 knots may be attributable to the malfunctioning roll sensor.

In general, the roll reduction of the BEAR's fin system as installed once all system repairs are made is considered to be at the lower bound of acceptable fin system performance. Due to the rather high vertical accelerations of this ship in the various crew work areas, however, this level of installed, active roll reduction capacity should be increased to the maximum extent feasible. Interference with manual crew tasks is directly related to both transverse acceleration (of which roll is the primary component) and normal acceleration magnitudes. By reducing transverse accelerations through roll stabilization, the effect is comparable in terms of task performance to reducing the bothersome normal accelerations. Thus, any added gain in roll reduction will serve to counter the crew performance degradation.

The initial sizing of the fins and the bilge keels for the BEAR was performed by DTNSRDC[10] in 1976 based on an assigned position of the fins on the ship. It is noted that this and similar work for other ships employed a rational helicopter and crew criteria-driven technique which used experimental, model roll damping for the hull, and theoretical processes for the lift developed by the fins.

Full-scale trial work in 1983 and 1984 including the BEAR and CONFIDENCE now suggests that there is serious doubt about the accuracy of the calculations by which fin and bilge keel size were determined due to the scaling inaccuracies between measured model and full-scale roll damping. These inaccuracies have resulted in an undersizing of the fins and bilge keels.

FIN SYSTEM REPAIR STATUS

Several deficiencies were noted as a result of the first two days of testing and were subsequently corrected during the time the BEAR was on its operational mission. The first problem was that the port and starboard fins did not attain the same maximum fin angle. Differences on the order of 4 degrees travel between the port and starboard fins were noted during the first two days of tests. This deficiency is typical of the type of minor correction generally made at the outset of fin evaluation trials.

It was reported and noted that the Speed Log input to the fin system was inoperative. This problem was traced to an improper rewiring of Speed Log Input in the IC switchboard. In the course of the modification to the pump stroke circuit for reducing the maximum fin rate, a loose pin was found, corrected, and determined to be the cause of the reported intermittent excessive port fin movement.

Since the correct speed input into the fin system is very important for the satisfactory performance of the fin system, Sperry Marine was requested to prepare a brief outline of the ship speed and fin angle calibration procedure. The resulting document which briefly outlines the theory of operation of the fin controller including the calibration was provided to the ship for inclusion in their manuals. This document should be formally forwarded to the remaining ships in the class and is therefore included as Appendix B.

Post trial analysis has indicated

1. that the ASU's roll angle sensor is inoperative and needs to be repaired;

2. that the fin system operational procedure is faulty in that the MANUAL GAIN rather than the AUTOMATIC GAIN mode was employed; and,

3. a revised simple operating instruction should be prepared for easy access by the operator.

SUMMARY OF CONCLUSIONS

It is to be noted that due to the repair status of the fins during the first trial, the results from the first three test sequences of this trial are considered to be somewhat less reliable measures of the true performance capability of the installed fin system than the fourth test sequence and the fifth one from the second sea trial. Furthermore, even these latter sea trial results understate the true performance capability of the installed fin system. This is particularly the case in quartering seas where the improper control law was employed due to the failure of the ASU roll sensor.

The use of the fin's MANUAL GAIN operating mode rather than the AUTOMATIC GAIN mode similarly did not permit the fin control system to select the proper overall system gains. Automatic gain adjustment prevents situations such as that measured during the first trial, where, in heavy seas, the fins operated in almost a "bang-bang" mode. This, in turn, introduces jerks into the roll and lateral response of the vessel and interferes with the crews' ability to properly self-stabilize themselves to the rolling motion of the ship.

Scaling inaccuracies between model-scale and full-scale roll damping have resulted in the installations of fins and bilge keels that are undersized for the ship. The correction of this undersizing is to employ the largest bilge keels and fins examined in the initial sizing calculations.[10] Details for the implementation of this correction are contained both in the text and Appendix A.

ACKNOWLEDGMENT

The authors are grateful to Mr. David A. Bennett of Sperry Corporation for his assistance in the BEAR sea trial data collection and his valuable contributions to this report.

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APPENDIX A

FIN SIZE UPGRADE DETAILS: BEAR CLASS

270-Foot WMEC Fin Upsize Project Outline

The practical utility of the simplest possible technique for increasing the moment generated by the fins was demonstrated to be feasible during the March 1984 seakeeping and fin stabilization trials with USCGC BEAR. The technique for increasing the moment-generating capacity of the fins was to increase the size of the fins without altering the installed machinery to actuate these fins.

During a discussion of the USCGC RRS program between DTNSRDC and USCG staffs on 7 November 1984, the fact that this upsizing of the fin has been proven feasible led to a suggestion by the USCG R&D sponsor of the RRS program. The suggestion was that a quick R&D project could establish the roll benefits of a fin with doubled capacity and the associated details of what this entails for the remaining twelve 270-ft WMEC's. DTNSRDC's position was that this is a relatively minor effort.

In order to verify this position, Sperry Marine was requested to discuss the details of such a quick project and to provide an approximation of the required effort to carry it out. This DTNSRDC request was passed to Sperry Marine during the 8 November 1984 meeting on the details of the proposed grooming of the four sets of fins on the existing 270's. The technical details of the Sperry response are included in this Appendix under the section entitled "Considerations for Increasing the Stabilizer Fin Size on the 270-Ft WMEC Class."

Fin System Alteration Details

The size of the new fins would require the construction of a new pair of fins with a 76" span and chord. The position of the shaft rotation axis in the fin would have to change slightly from its geosim location to lower the static torque required to position the fin. It is to be recognized that the static torque component represents a dominant term in the total torque required to move the fin in accordance with the existing control algorithm.

The repositioning of the shaft axis within the fin is quite important in order to limit the horsepower required to move the fin at one-half of the present maximum fin rotation rate of 42 degrees per second. It is to be noted that there are no costs associated with this repositioning of the shaft axis in the fin because a new set f fins has t e uilt in any case. Thus the change in the location of the fin shaft axis is of no importance in the manufacturing process.

The reduction of the maximum fin rate will permit the use of existing fin pump power unit and associated hydraulics. However, the size of the hydraulic cylinders needs to be increased for the actuation of the fins.

The cylinders internal swept volume has to be increased by a factor of two in order to reduce the maximum fin rate by a factor of two without changing the fin stroke. This reduction in the fin rate permits the power available to move the fin through the existing maximum fin angle excursion. No changes in the control algorithm are required.

Since the cylinder actuating the larger fin now transmits essentially twice the force as for the smaller fin, the path of this load back into the ship has to be strong enough to withstand this load. Similarly, the local structure where the cylinder is attached to the hull may similarly need to be strengthened. The design and construction of the larger cylinder will provide a stronger load path such that it will only require the rotating knuckles and perhaps the tiller to be strengthened. None of these items are considered to be of much cost consequence.

The transmittal of the increased fin force into the hull at the shaft-to-hull stave bearing may require increasing the wear resistance of the bearing. This can be achieved by changing the sleeve material from its present gun metal composition on the first four ships to the monel material used on the last nine ships as per the experience gained by the manufacturer during the accelerated wear test of the FFG 7 fin systems at DTNSRDC Annapolis.

Additional details which explain the proposed fin increase are included as the last section in this Appendix.

Cost Estimate

Sperry Marine was requested informally on 8 November 1984 to provide a cost estimate to be used for program planning purposes. The results of this estimate are hereby included in this text for the sake of completeness.

The price of the 25 square foot fins currently installed in the BEAR class was quoted to be $13,000 per fin when these were purchased as part of the 13 fin set. These fins were manufactured evidently in the U.S. under a subcontract to Sperry Marine. In order to build one set of fins of increased size (i.e., to the 40 square foot dimension and new shaft axis of rotation) requires:

1.	Developing of new Casting Pattern	$31.0K
2.	Casting	$11.0K
3.	Fabrication	$21.0K

These very large costs for the upsized BEAR fins led to the investigation of costing the building of these fins in Japan. Within a four to five month period upon receive of funding, a substantial overall cost saving could be achieved by building the fins there. Thus, the total resulting Sperry cost estimate for the upsized fins is as follows:

1.	Fin Set	$44.0K
2.	Shaft Sleeves, Monel per set of 2	$12.0K
3.	Cylinder set, 4 per set incl rods & ends	$12.0K
4.	Engineeering and design	$15.0K
	Total Sperry Cost	$83.0K

This total represents the initial cost for the first upsized fin set and a reduction can be expected for follow-on production. However, it is to be noted that these costs do not represent the total cost to USCG for the upsized fins. The costs associated with the installation of these fins on the ship is the major remaining cost not accounted for. This undefined cost is considered to consist primarily of the cost for drydocking the ship; removal of the existing fins (in the case of the first four BEAR class cutters, the possible removal of the fin shaft and installation of monel sleeves, if necessary); installation of new fins; and at-sea testing of the fins for performance. These limited at-sea trials could, of course, be conducted by the USCG instructors in the presence of a single DTNSRDC engineer with equipment furnished as part of the fin grooming and training project.

A substantial uncertainty remains in the cost associated with the installation of the upsized fins. This uncertainty is largely the result of timing such work on each cutter. Should such work be conducted as part of other work in progress which already requires the drydocking of the vessel, the total installation cost would clearly be reduced. Thus, should fin upsizing be performed, for example, on one of the latter ships in the class still under construction, the installation costs can be essentially avoided since the larger fins would be installed in place of the smaller ones.

Once the initial ship with upsized fins has been found to be satisfactory, the cost of the modifications of the subsequent 12 ships should be substantially lower than the cost for this first vessel.

It is to be pointed out that the upsizing of the BEAR fins is expected to increase the roll reduction of the BEAR class cutter at 15 knots in a Sea State 5 by 15 to 30 percent based on simulation results for the PCG, the PGG, and the PCM. Furthermore, this increased level of roll reduction performance can be achieved at a cost of about one-twelfth of the cost of the fins as obtained from Sperry Marine at the outset. It is evident from these figures that no alternative method for increasing the roll reduction performance of the BEAR class, including RRS systems of the type currently designed and under development, are as cost effective.

CONSIDERATIONS FOR INCREASING THE STABILIZER FIN SIZE ON THE 270-FT WMEC CLASS*

The following text considers the feasibility of improving the effectiveness of the roll stabilizers on the 270-ft WMEC vessels by increasing the size of the stabilizer fins within the constraints imposed by the ship's block outline, utilization of the existing fin stock and supporting structure, and requiring no increase in the level of installed hydraulic power.

Previous studies using hybrid simulation techniques with special attention to modeling of the stabilizer fin and its effect upon the ship for PGG, PCG, PCM, and 270-ft WMEC vessels have all indicated that fin rotational rates can be reduced from normal specification practice without significant penalty in the roll reduction performance.

The simulation models have been verified by at-sea tests of the PCG, PGG, and 270-ft WMEC class vessels. The effect of fin rate capability reduction was tested on WMEC-901 in March 1984. Results were consistent with the simulations.

Previous studies have also shown that the fin stock and associated machinery is capable of supporting a much greater load than currently developed in the 270-ft WMEC stabilizers.

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*Prepared by D.A. Bennett, Engineering Staff Consultant, Sperry Corporation Aerospace and Marine Division.

Existing fin blades are fitted onto a tapered fin stock and could be removed by drawing off from the taper. Larger sized fin blades could then be fitted onto the same tapered stock.

Based upon the above indictions, it is considered feasible to increase the size of the fins on the 270-ft WMEC vessels from 25 square feet to about 40 square feet, operating the fins with a reduced turn rate capability consistent with the currently installed hydraulic power, and achieving significant improvement in the roll motion attenuation at all vessel speeds.

Fin simulation work actually took the fin rates down to values of 16.8 degrees per scond and still indicated that there was essentailly no degradation in the roll reduction achieved by the fins.

One issue of the increase in the fin loads not previously mentioned are the pressure fluctuations in the fin cylinders produced by the buffeting due to the local wave effects and the potential of approaching the water surface or even emerging therefrom. These forces were found to be of substantial magnitude but are considered to be of values that can be accommodated by the existing machinery without modifications. It is to be noted, however, that the validity of this latter preliminary examination needs to be checked in some detail as part of the design engineering work for the first ship.

Based on a maximum fin rate of 16.8 degrees per second, the increases in the stress and load factors due to the increased fin size are given in the following table:

Load and Fin Particulars	25 sq ft	40 sq ft	Ratio 40/25
Peak Static Torque (lb-ft)	5786	6880	1.19
Peak Rate Torque (lb-ft)	9264	9402	1.015
Peak Total Torque (lb-ft)	10923	11650	1.067
Max Fin Lift (tons)	6.16	10.92	1.77
Max Total Force (tons)	6.28	11.19	1.78
Max Bending Moment Ratio	1.00	2.26	2.26
Fin Area (sq ft)	25	40	1.60
Buffeting Loads, Bending	1.00	2.026	2.026
Buffeting Loads, Torsion	1.00	2.026	2.026

All of the load factor increases are within the reserve factors applied in the stress calculations so it seems certain that the fundamental strengths of the shaft and associated parts will be adequate. The increases in the buffeting loads in torsion will reduce the fatigue reserves and will probably require some strengthening of the tiller attachment, tiller key, hydraulic cylinder supports and the cylinder rod-end bearing.

In order to utilize the same motor/pump assembly, the reduced fin rotational rate is probably best accomplished by increasing the area of the actuating cylinders. This will be accompanied by the stronger cylinder rods, rod-ends and bearings. It may also be advantageous to increase the tiller radius a little in order to accommodate beefier rod-end bearings.

The increase in bending moment by a factor of 2.26 is probably the major area for investigation into suitability of the existing bearing materials. The outer bearing is a "stave" bearing, phenolic-resin staves for the outer of the cylindrical bearing and a sleeve on the shaft forms the inner.

The first four pairs of stabilizers for the 270-ft WMEC vessels incorporated a gun metal sleeve on the shaft, the design being based on the U.S. Navy specifications for prototype equipment for the FFG-7 class, in which the sleeve, shrunk-on, was a requirement, in a choice of copper alloy or monel material.

Accelerated wear tests on the FFG-7 prototype fin system at DTNSRDC, Annapolis, showed that monel sleeve material had better wear properties than the gun metal, at the high loading levels used in the tests.

In the subsequent manufacture of the remaining nine ships' sets of stabilizers for the WMEC, the shaft sleeve was made of monel metal because of its superior demonstrated wear properties.

At the existing 25 square foot fin size, the original gun metal sleeves should have an adequate life capacity, but with a bearing load increased by 2.26 times it must be recommended that the shaft sleeves should be replaced by monel metal sleeves.

When the increased fin size is tried out on one ship--to be evaluated by seakeeping fin performance tests and general service experience, the suitability of the existing bearing and sleeve materials for use with the larger 40 square foot fins could be part of that evaluation. The details of such a program can be furnished upon request.

Bearing Material Suitability for Larger BEAR Fins

If the increased fin size is to be tried out on one ship--to be evaluated by seakeeping tests and general service experience, the suitability of the existing bearing and sleeve materials for use with the larger sized fins could be a part of that evaluation.

Replacing one fin blade by another does not require removal of the fin shaft.

Removal of the fin shaft requires dismantling of the tiller and inner bearing assembly. In replacing a shaft sleeve--machine off the old and shrink-on the new--it is the removal and re-assembly which will incur most of the cost and time. Thus if a shaft from one of the first four BEAR class cutters is to be withdrawn, it makes economic sense to replace the sleeve at that time.

On the first ship therefore, the options in ascending order of cost are:

a. leave the existing shafts and sleeves in position

b. withdraw one shaft, inspect the bearing and sleeve (as indicative of the state of the other) then fit a new monel sleeve and re-assemble

c. withdraw both shafts and fit new monel sleeves to both.

APPENDIX B

OPERATION/CALIBRATION OF BEAR CLASS FIN CONTROLLER*

*Prepared by D.A. Bennett, Engineering Staff Consultant, Sperry Corporation Aerospace and Marine Division

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21 March 1984

David Taylor Naval Ship Research
And Development Center
Ship Performance Department
Bethesda, Maryland 20084

ATTN: Mr. T. R. Applebee

REF: Fin Calibrations on WMEC-270 Class

Dear Terry:

Our recent experience on USCG Cutter Bear indicated that the speed dependent calibration of the fin angle limits may not be too well explained or understood.

The attached note is offered to explain and define the calibration aims and procedure.

Your comments are invited.

Very truly yours,

SPERRY CORPORATION

D A Bennett

D. A. Bennett
Engineering Staff
Consultant

:fr

cc: N. Addington
E. Rosson
W. Spurgin

Attachment

THE GYROFIN STABILIZER CONTROL SYSTEM SHIP SPEED AND FIN ANGLE CALIBRATION

1. Design studies prior to building, select the fin size and shape, and the speed of ship at which the fins are to have their maximum effect. (The "Design Speed").

   At speeds above the Design Speed, the maximum lift force generated by the fins is required to remain constant - a structural strength consideration.

   At speeds below the Design Speed the maximum fin angle is required to remain constant, at a level which avoids stalling or excessive cavitation.

2. Basic ship parameters of interest are given in Table 1, and the calculated (required) fin angles and fin lift forces at various speeds are given in Table 2. The required maximum values of fin angle, and fin lift are shown graphically in Figure 1.

   In selecting the calibrated maximum values, allowance has been made for the fact that residual roll motions of the vessel in rough seas will cause an increase in the effective angle of the fin to the local water stream direction.

   The lift coefficients and lift force values in Table 2 apply in steady-state conditions of ship speed and zero roll rate.

3. The lift force generated by the fin may be calculated from:

   $$\text{Lift} = \text{Fin Area } (\text{ft}^2) \times .043\ \beta \times (V \times 1.689)^2 \text{ lbs.}$$

   which reduces (for WMEC 270 class ships) to:

   $$\underline{\text{Lift} = 0.00137\ \beta . V^2 . \text{ Long Tons.}}$$

   where: β = Fin Angle (Degrees)
   V = Ship Speed (Knots)

4. The Gyrofin Control System incorporates a Lift Order Computer, which uses data from ships roll motion sensors to calculate, instant by instant, the lift required from each fin to counteract the rolling motions of the vessel.

   The Lift Order Computer amplifier is followed by a Lift Order Limiter Stage which imposes the Lift Order limits shown on Figure 1.

   The output voltage (Limited Lift Order Volts) is scaled such that the amplifier maximum output volts (about 10 volts) represents the maximum

4. Lift of 6.16 Tons. As can be seen from the equation of paragraph 3, if the maximum Fin Angle is to remain constant at speeds below the Design Speed, the maximum Lift Order must be reduced proportionally to Speed squared (V^2). This is achieved by supplying a voltage proportional to V^2 to the Lift Order Limiter Circuit.

5. The Fin quantity measured and available for control purposes, (in the case of WMEC 270 Class Ships) is Fin Angle. (In some other ships the quantity Fin Lift is also available). So when the Lift Order signal is sent to the fin servo controls in each local Control Unit, it must be translated into a Fin Angle order signal for comparison with the achieved Fin Angle quantity. The translation is accompished in the Stroke Order Computer where the first amplifier, AR1.D, has the multiplier U1, in its feedback path, with V^2 volts as one of the inputs to the multiplier and AR1.D output as the other input. The effect of this is to cause the output of AR1.D to be ...(constant x input x $1/V^2$) which is equivalent to the desired Fin Angle Command.

6. The equation in paragraph 3 can also be interpreted as Lift per unit of Fin Angle = constant x V^2. This "Lift per Unit Angle" is one of the gain parameters of the stabilizers and since it varies with V^2, the quantity V^2 is used as the reference voltage for the Automatic Gain Control Circuit.

7. The three functions described in paragraphs 4, 5 and 6 each require a voltage proportional to the square of speed, i.e. V^2. This is generated in the Computer Pre-amplifier board, (Part of the Analog Processor Unit), unit 1A1.A3.

The circuit diagram for V^2 generation is shown in Figure 2 from which it can be seen that ship speed information may be obtained from the ship's speed log via the local I.C. Switchboard, or be set in manually by calibrated potentiometer on the master control panel.

8. When Auto Speed is selected, a negative D.C. speed dependent voltage is taken from the Auto Speed calibrated potentiometer to the amplifier AR1D where it is sign inverted without scale change and then applied to the manual, Auto Switch.

Manual speed voltage is derived from the calibrated manual speed potentiometer on the master control panel, which is supplied with +15V D.C. from the computer D.C. power supply (1A1.A1).

The selected voltage is then applied to the amplifier AR2B, and then to the multiplier via diodes which prevent reverse speed voltage from being applied.

9. V^2 Calibration - A standard calibration procedure is used which requires that the V^2 voltage from the multiplier will be 6.2V DC at the "Design" speed for the vessel.

In the case of WMEC-270 class ships, the Design Speed is 15 Knots so the calibration procedure is to have the ship at 15 knots, and to set the manual speed knob at 15 knots, then to adjust either Auto Speed Cal, or Manual Speed Cal, as selected, to achieve 6.2V at TP J1-P.

10. Fin Angle Calibration

The objective in calibrating the fin angle is to ensure that maximum level commands from the Lift Order Computer result in maximum Fin Angles as shown on Figure 1.

Lift Order Signals are subjected to limiting as a function of ship speed in the Lift Order Limiter stage 1A1.A6, and to translation into Fin Angle Commands (also as a function of ship speed) in the Stroke Order Computer, amplifier AR1.D.

The Fin Angle which satisfies the resulting command is controlled by the Angle Calibration Control, R5 in each Lift and Fin Angle Amplifier (in the Local Control Units).

Fin Angle calibration therefore requires that the speed dependent stage, V^2, is first set correctly. (See paragraph 9). An overall check of the calibration may then be made by operating the Forced Roll switch on the Master Control Unit and then adjusting the Angle Calibration Control R5 to obtain the desired maximum Fin Angle. The procedure requires two men, one at the Master Control Unit, and the other at the Local Control Unit, with communication between the two. Each fin should be checked, in each direction.

11. Recalibration After Unit Replacement

Replacement circuit cards will not normally have their controls adjusted to the specific ship values prior to installation, and component values may vary slightly from one board to another within the allowable tolerance band. Recalibration of the V^2 voltage and the Fin Angle, as defined in paragraphs 9 and 10, should therefore be carried out if any of the following circuit cards are replaced:

(a) Power Supply 1A1.A1, 4(P)A1, or 4(S)A1.

(b) Computer Preamplifier 1A1.A3

(c) Lift Order Limiter 1A1.A6

(d) Lift and Fin Angle Amplifier 4(P)A2 and 4(S)A2.

(e) Stroke Order Computer 4(P)A3 and 4(S)A3.

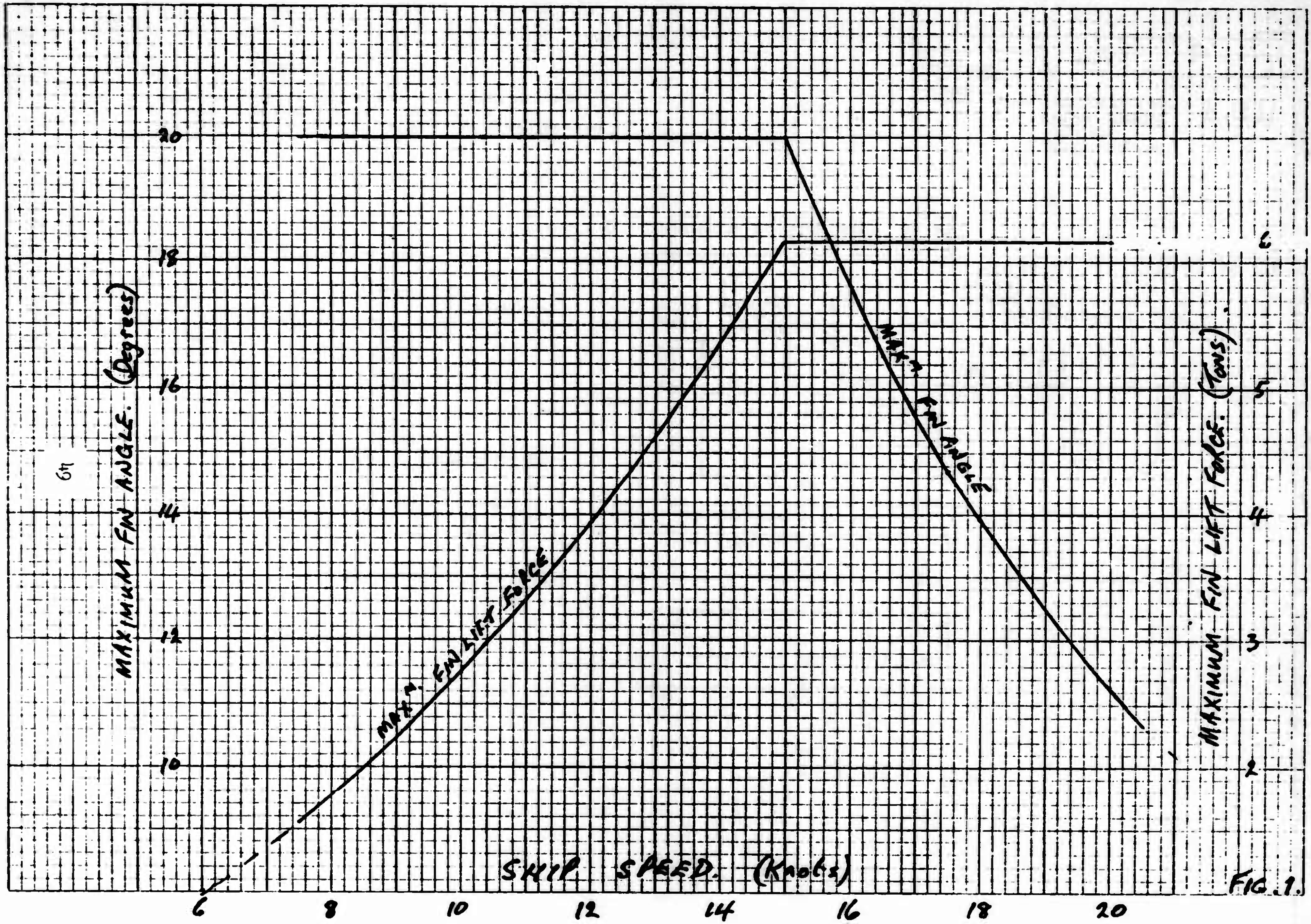
Table 1.

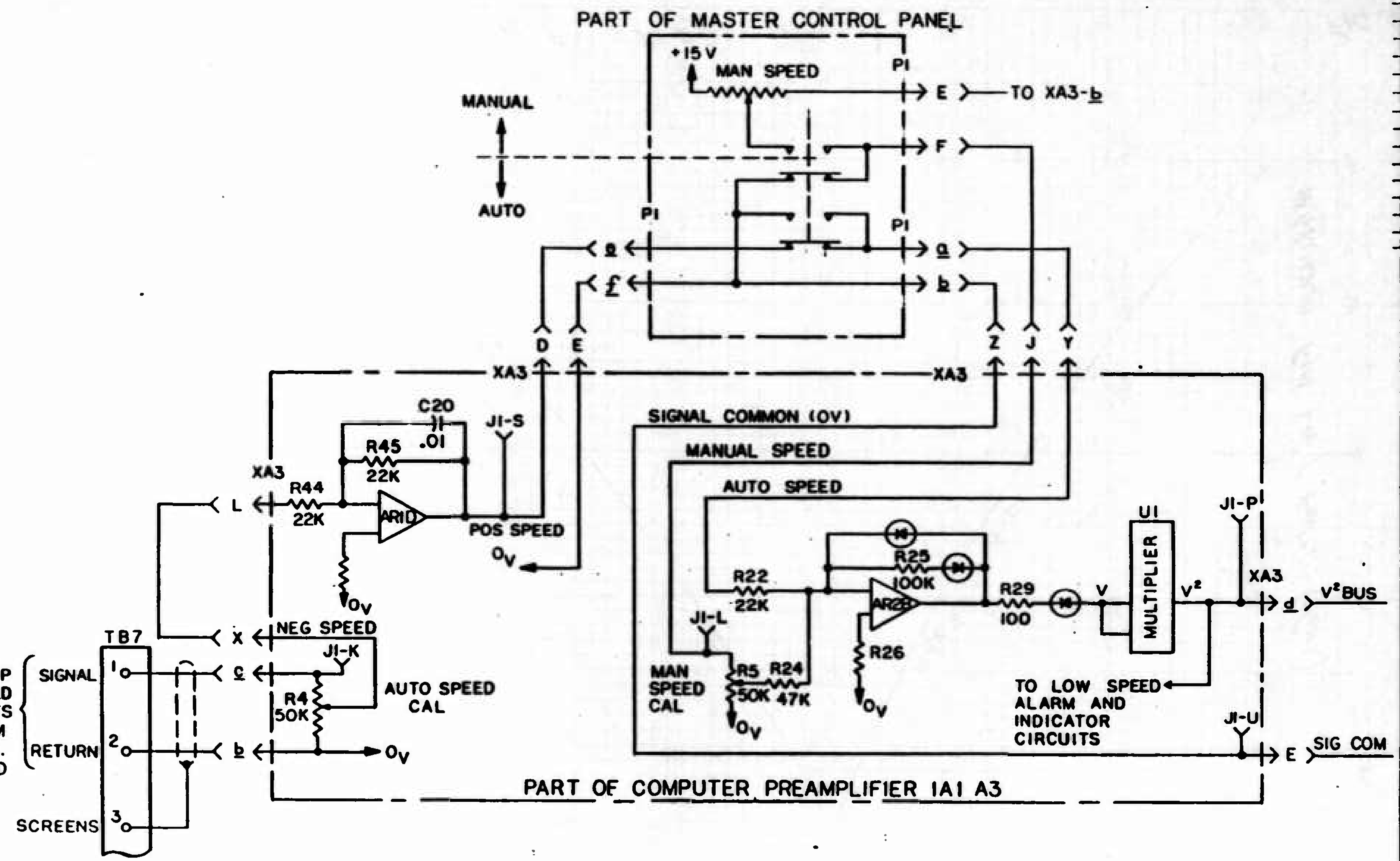
WMEC 270 CLASS SHIP/STABILIZER DATA

LENGTH Lpp.	255 ft.
DISPLACEMENT (FULL)	1768 LONG TONS
MAX. BEAM	38 FT.
DRAFT	13.5 FT.
METACENTRIC HEIGHT (GM)	2.18 FT.
VERTICAL CG (KG)	17.74 FT.
STABILIZER DESIGN SPEED	15 KNOTS
NATURAL ROLL PERIOD	11.2 SECONDS
FINS: 2 x 25 SQ. FT. SECTION NACA 0015 GEOMETRIC ASPECT RATIO 1.0 STALL ANGLE APPROX. 24 DEGREES	

Table 2

SHIP SPEED (KNOTS)	7.5	10	12	15	17.5	20
MAX FIN ANGLE (DEGREES)	20	20	20	20	14.7	11.25
LIFT COEFFICIENT CL	0.86	0.86	0.86	0.86	0.63	0.48
MAX LIFT FORCE (TONS)	1.54	2.74	3.94	6.16	6.16	6.16


MAXIMUM FIN ANGLE. (Degrees)
MAXIMUM FIN LIFT FORCE. (Tons)
SHIP SPEED. (Knots)
MAXm. FIN LIFT FORCE
MAXm. FIN ANGLE
20
18
16
14
12
10
6
5
4
3
2
6
8
10
12
14
16
18
20
FIG. 1.



V^2 BUS CONNECTED TO:
(a) LIFT ORDER AGC BOARD 1A1 A4 PIN r. TEST POINT J1-O (AGC REFERENCE VOLTS).
(b) LIFT ORDER LIMITER BOARD 1A1 A6 PIN B. TEST POINT J1-A.
(c) STROKE ORDER COMPUTERS 4 (P) A3 AND 4 (S) A3 PIN A. TEST POINT J1-E.

GYROFIN STABILIZER (SHIP SPEED)^2 VOLTAGE GENERATION (MEC 270 CLASS SHIPS)

SPERRY

C 03956

FIG. II

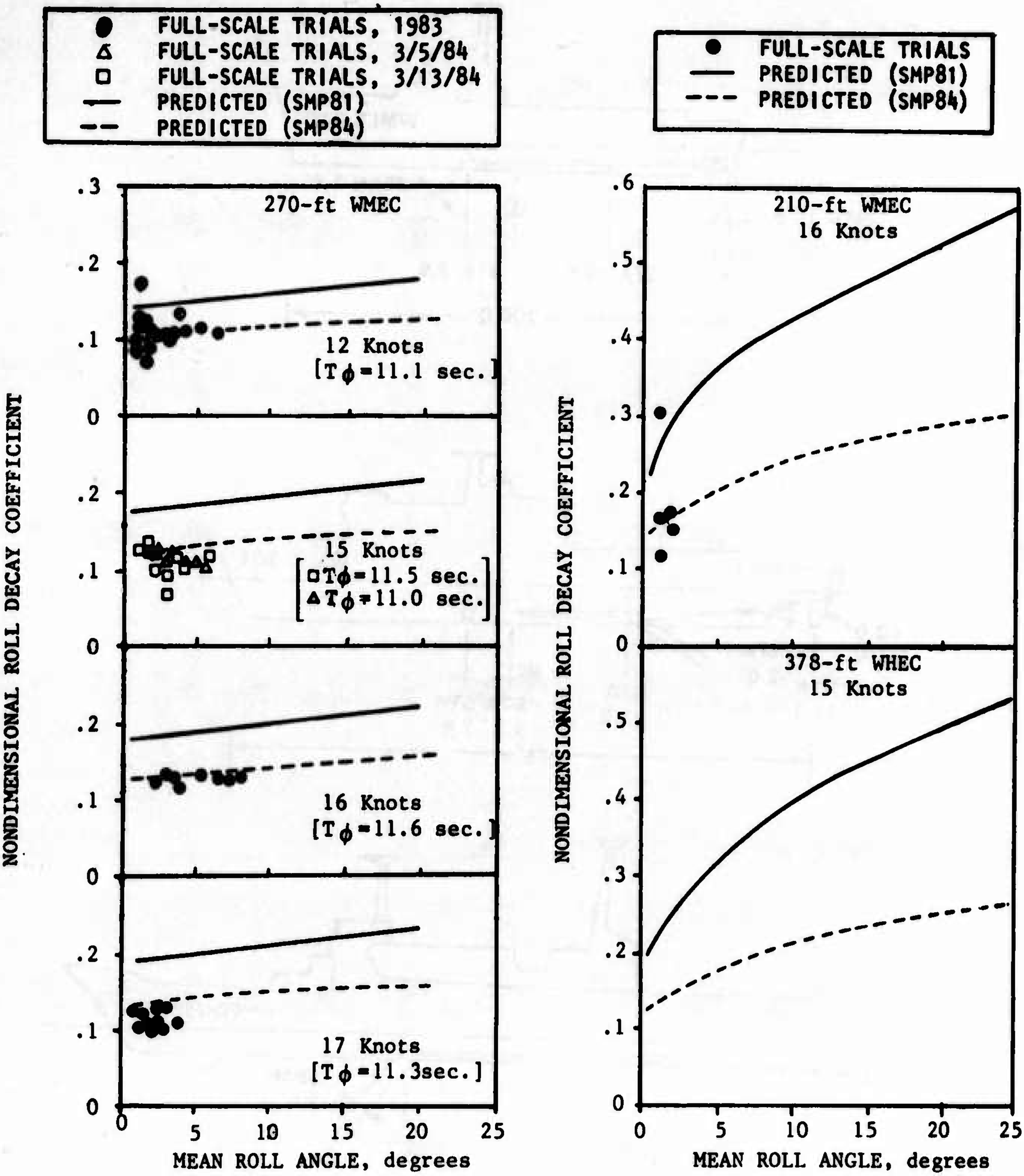


Figure 1 - Comparison of Measured and Calculated Roll Damping for the WHEC-715, WMEC-901 and WMEC-615 Cutters. Calculations with SMP-81 and SMP-84

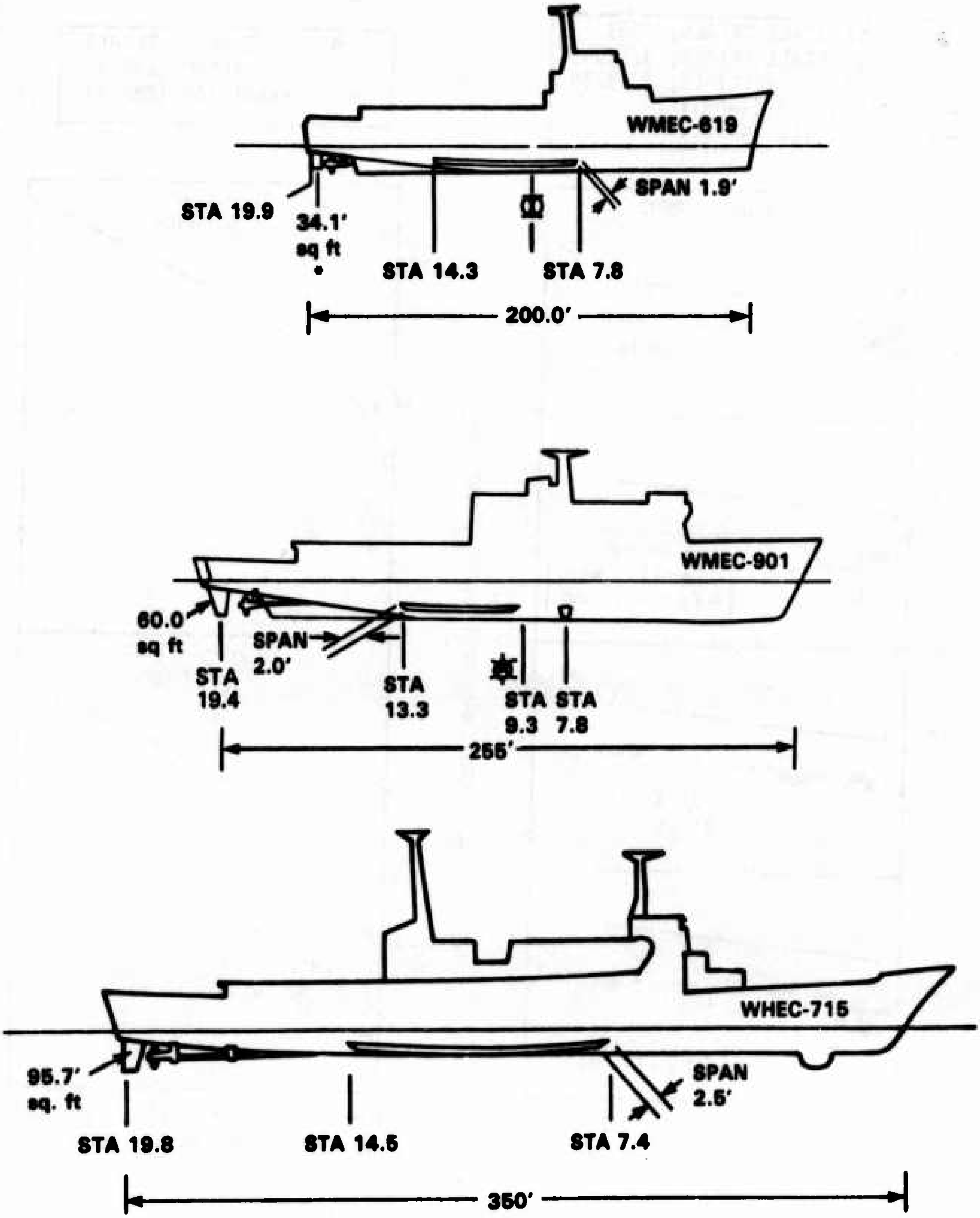


Figure 2 - Profile of WHEC-715, WMEC-901 and WMEC-615

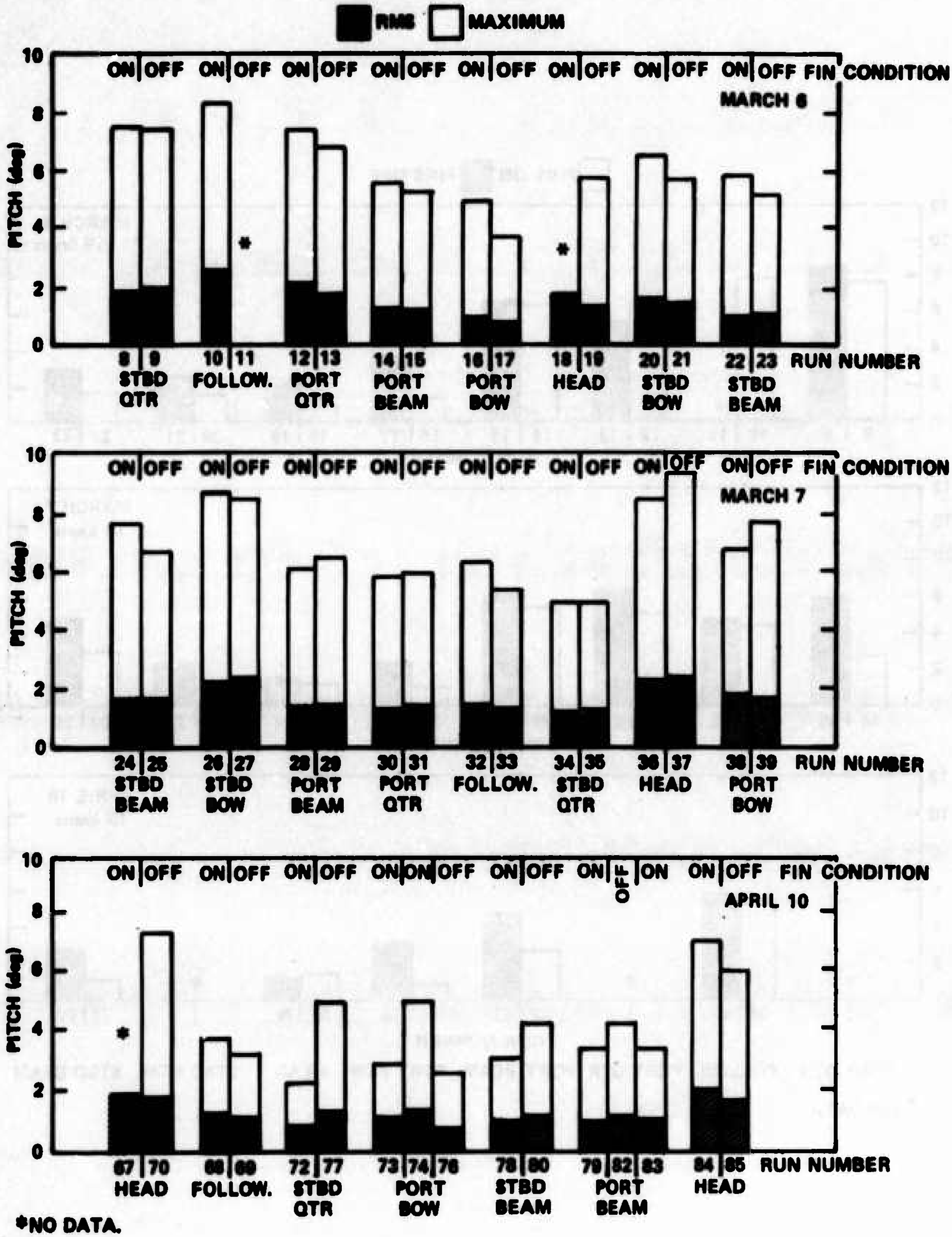


Figure 3 - Effect of Fin Activity on Ship Pitch Motion

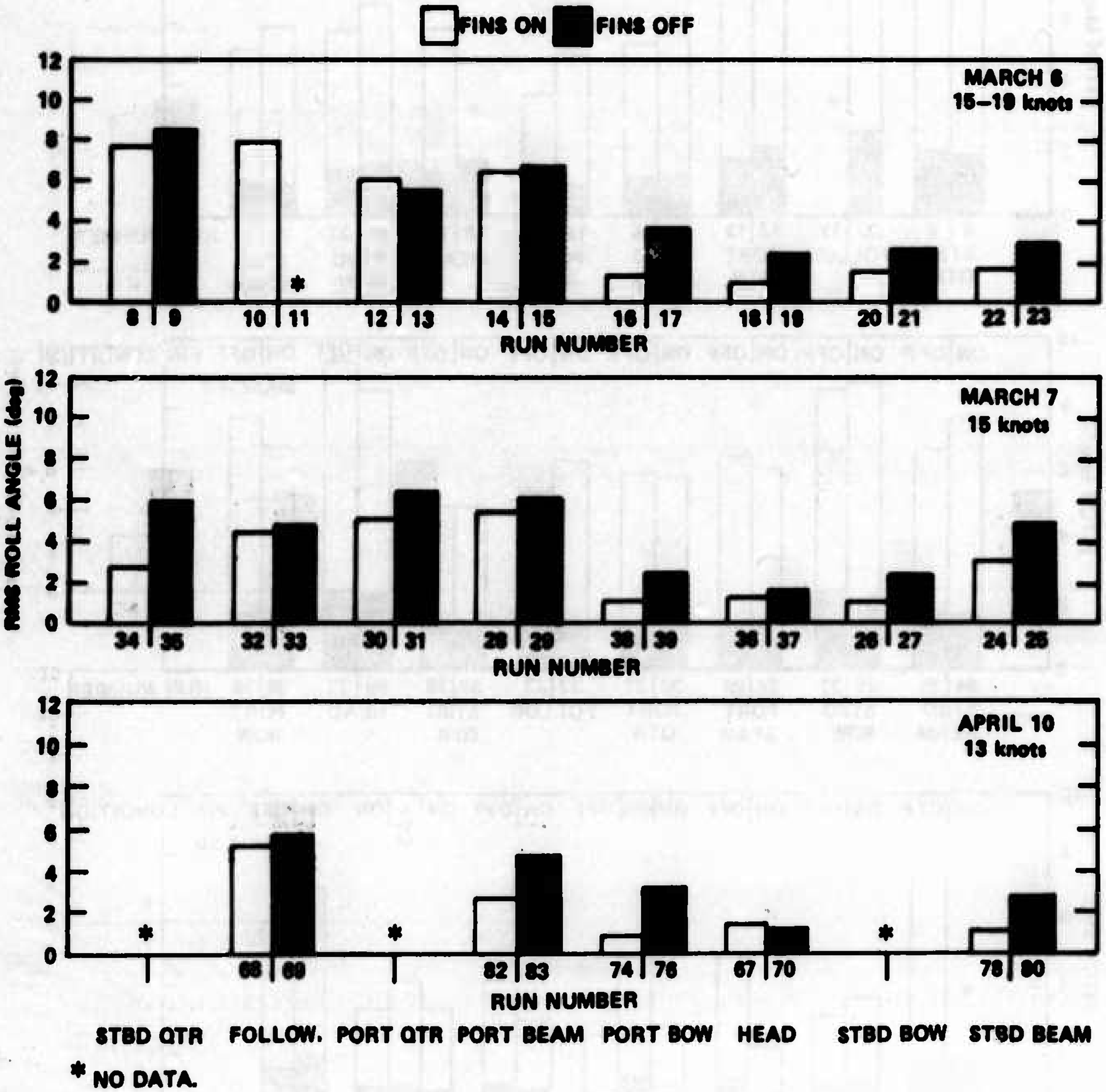


Figure 4 - Summary of Measured WMEC-901 BEAR Fin Stabilizer Performance

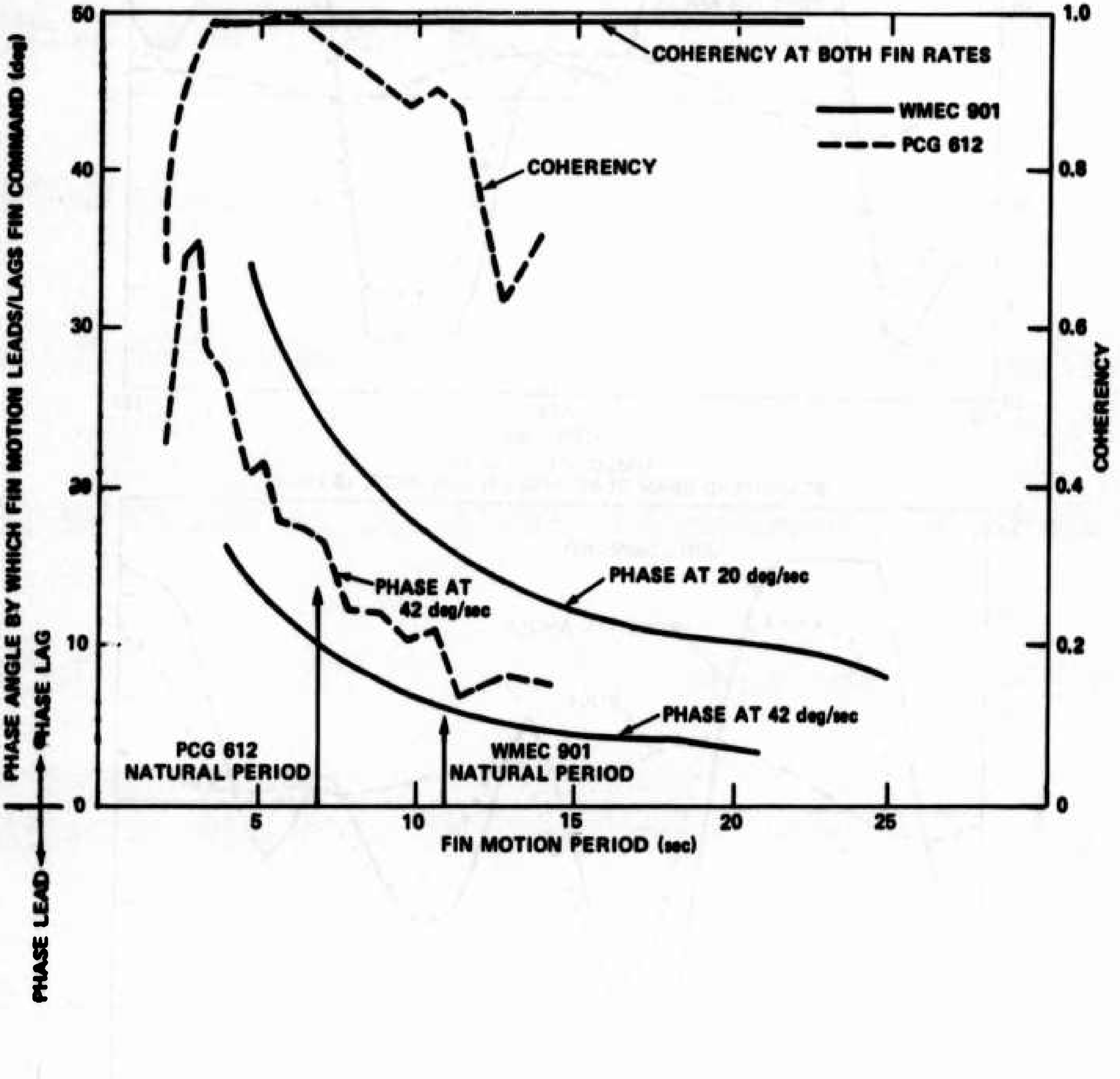


Figure 5 - Fin Machinery Performance in Following the Stabilizing Fin Command Signal for WMEC-901 and PCG-612

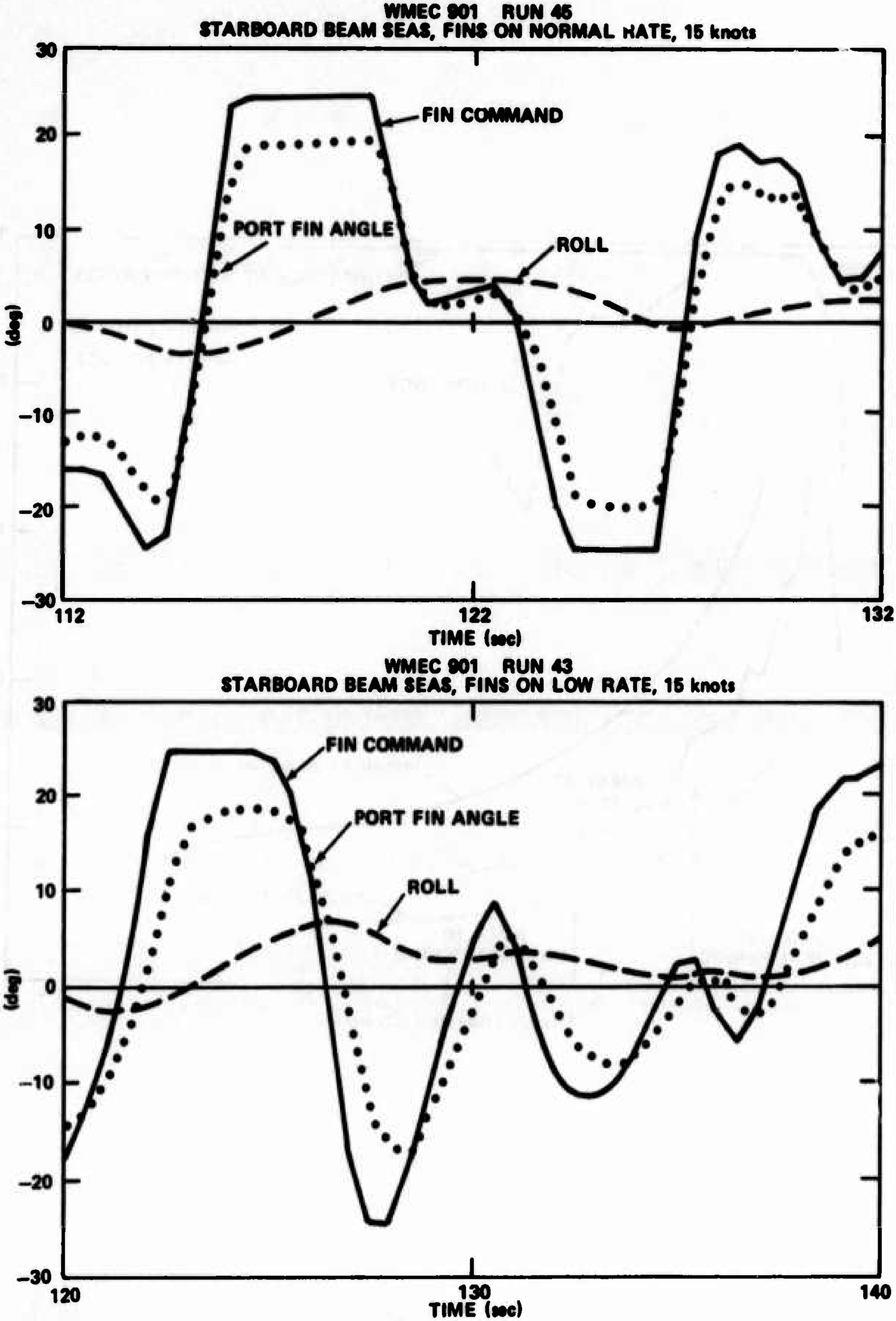


Figure 6 - Time Histories of Fin Machinery Performance at Normal and Low Fin Rates

Figure 7 - Maximum Fin Angle and Rate Reduction Effects on Roll Stabilizing Performance

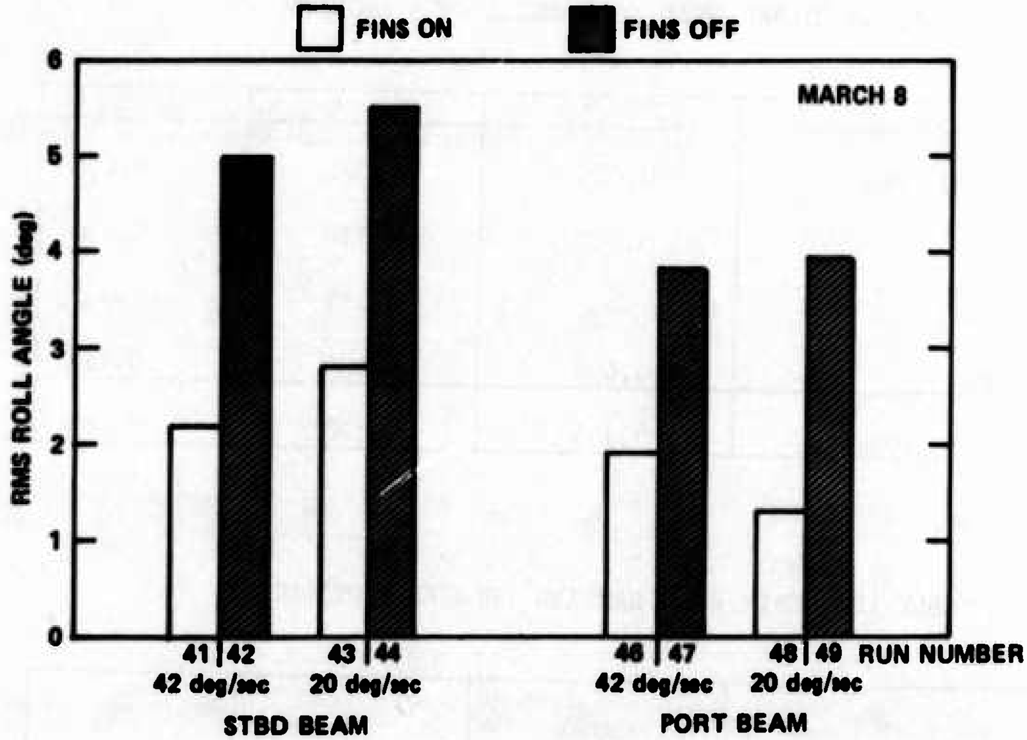


Figure 7a - Fin Rate Reduction Effect on Fin Performance

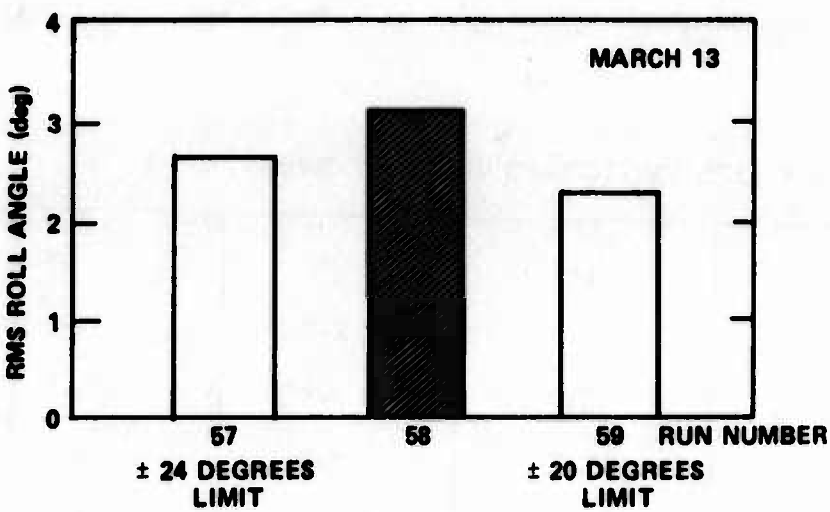


Figure 7b - Reduction of Fin Angle Limit Effect

TABLE 1 - ROLL DAMPING, SHIP AND BILGE KEEL PARTICULARS FOR THE 378-FT, 270-FT and 210-FT USCG CUTTERS

TABLE 1A - NONDIMENSIONAL ROLL DAMPING COMPONENTS FOR A SHIP SPEED OF 15 KNOTS AND MEAN ROLL ANGLE OF 5 DEGREES

	378-ft	270-ft	210-ft
Bare Hull and Skeg	0.027	0.041	0.053
Rudders	0.050	0.032	0.036
Fins	---	0.031	---
Bilge Keels	0.098	0.025	0.114
Total	0.174	0.128	0.202

TABLE 1B - SHIP ROLL DAMPING RELATED PARTICULARS

T_ϕ (roll period) (sec)	10.6	11.2	11.4
GM (ft)	2.93	3.1	2.0
Displacement (L.T.)	3016	1790	1009

TABLE 1C - BILGE KEEL PARTICULARS INCLUDING MOMENT ARM Y

Length (ft)	124	52	65
Span (ft)	2.5	2.0	1.93
Area (sq ft)	310	104	125.5
Moment Arm, y (ft)	22.3	18.3	18.0
Longitudinal Location (ship stations)	10 - 14	10 - 13	8 - 14

TABLE 2 - EFFECT OF INCREASED BILGE KEEL AND FIN SIZE ON ROLL DAMPING AND MOTIONS OF THE WMEC-901

TABLE 2A - NONDIMENSIONAL ROLL DAMPING COMPONENTS FOR A SHIP SPEED OF 15 KNOTS AND MEAN ROLL ANGLE OF 5 DEGREES

	As Is	Increased BK	Increased BK + Fin
Bare Hull and Skeg	0.041	0.041	0.041
Rudders	0.032	0.032	0.032
Fins	0.031	0.031	0.051
Bilge Keels	0.025	0.040	0.040
Total	0.128	0.144	0.165

TABLE 2B - UNSTABILIZED AND STABILIZED RMS ROLL IN 13-FT SIGNIFICANT WAVE HEIGHT, 9-SECOND MODAL PERIOD SHORTCRESTED BEAM SEAS

Unstabilized RMS Roll (deg)	4.2/4.8	4.0/4.5	3.9/4.2
Stabilized RMS Roll (deg)	N.A./2.9	N.A./3.0	N.A./1.7
Definition: SMP81 Roll/SMP84 Roll; N.A. = Capability not available.			

TABLE 2C - BILGE KEEL AND FIN PARTICULARS

BK Length (ft)	52	52	52
BK Span (ft)	2.0	3.0	3.0
BK Area (sq ft)	104	156	156
BK Moment Arm, y (sq ft)	18.3	18.6	18.6
BK location, ship stations	10 - 13	10 - 13	10 - 13
Fin Mean Chord (ft)	5.0	5.0	6.325
Fin Mean Span (ft)	5.0	5.0	6.325
Fin Area (sq ft)	25	25	40
Fin Moment Arm, y (sq ft)	19.86	19.86	19.93
Fin Location, ship station	7.68	7.68	7.68

TABLE 3 - MACHINERY PERFORMANCE POWER SPECTRA FOR WMEC-901

Run Number 45, Trial Number 1, March 1984
STBD Beam Seas (5 to 6 ft), 15 Knots, Sperry Marine Fins ON
Max Fin Rate = Normal Nominal 42 deg/sec
ISKIP=2; Filter = 0.5 Hz; 64 Frequency FFT
POWER SPECTRA: S1 = Fin Command; S2 = Port Fin Angle

Enc Period T_{OE} (sec)	Enc Freq ω_e (rad/sec)	S1 Spectral Ordinates (deg^2-sec)/rad	S2	Phase (S2 - S1) (deg)	Coherency dimensionless
128.0	0.049	6.910	3.927	1.9	0.987
64.0	0.098	8.150	4.795	0.3	0.986
42.7	0.147	16.932	10.200	-1.4	0.997
32.0	0.196	25.606	16.138	-1.8	0.997
25.6	0.245	48.597	30.591	-2.7	0.998
21.3	0.295	108.914	69.343	-3.3	0.999
18.3	0.344	315.703	199.308	-4.0	1.000
16.0	0.393	587.488	369.309	-4.3	1.000
Peak →14.2	0.442	637.754	397.519	-4.9	1.000
12.8	0.491	609.207	380.961	-5.5	1.000
11.6	0.542	404.228	254.414	-5.6	1.000
10.7	0.587	159.941	100.224	-6.5	0.999
9.8	0.641	164.463	103.978	-7.0	0.999
9.1	0.690	184.473	115.987	-7.0	0.999
8.5	0.739	114.041	70.586	-8.0	0.999
8.0	0.785	153.256	94.478	-8.8	0.999
7.5	0.838	84.153	51.963	-9.7	0.999
7.1	0.885	61.035	37.065	-10.2	0.998
6.7	0.938	104.440	63.327	-10.0	0.998
6.4	0.982	101.094	62.688	-10.4	0.998

Run Number 43, Trial Number 1, March 1984
Same conditions as Run Number 45, except Max Fin Rate = 20 deg/sec

32.0	0.196	30.954	17.770	-4.1	0.995
25.6	0.245	39.983	22.235	-7.9	0.997
21.3	0.295	83.563	47.059	-10.0	0.997
18.3	0.344	328.416	181.827	-10.6	0.999
Peak →16.0	0.393	575.788	312.595	-11.5	0.999
14.2	0.442	530.929	286.723	-12.4	0.999
12.8	0.491	527.859	285.492	-13.7	0.999
11.6	0.542	385.898	205.039	-15.6	0.998
10.7	0.587	307.290	162.991	-17.5	0.998
9.8	0.641	230.989	124.508	-18.0	0.999
9.1	0.690	138.685	74.821	-19.6	0.998
8.5	0.739	99.322	51.505	-20.9	0.997
8.0	0.785	83.441	42.590	-22.5	0.997
7.5	0.838	112.490	57.025	-23.1	0.999
7.1	0.885	117.242	57.515	-24.0	0.998
6.7	0.938	101.693	48.825	-25.6	0.995
6.4	0.982	115.373	54.654	-27.4	0.996

TABLE 4 - MACHINERY PERFORMANCE POWER SPECTRA FOR WMEC-615 and PCG-612 CLASSES

Run Number 79, CONFIDENCE 1984 Sea Trial with RRS
RRS Active, 15 Knots; 7-8 Ft Quartering Seas
ISKIP=3; Filter = 0.5 cps

Enc Period T_{OE} (sec)	Enc Freq ω_e (rad/sec)	S1	S2	Phase (S2 - S1) (deg)	Coherency dimensionless
		Spectral Ordinates (deg^2-sec)/rad			
60.0	0.105	58.173	55.754	-5.2	0.982
Peak →40.0	0.157	105.749	97.187	-3.8	0.986
30.0	0.209	63.380	56.807	-5.4	0.969
24.0	0.262	49.812	47.415	-5.2	0.965
20.0	0.314	73.462	69.566	-6.0	0.981
17.1	0.367	69.469	62.726	-7.7	0.979
15.0	0.419	61.724	54.682	-9.2	0.972
13.3	0.471	29.729	27.490	-9.4	0.955
12.0	0.524	25.178	20.695	-11.3	0.950
10.9	0.576	25.841	21.583	-11.4	0.945
10.0	0.628	19.520	15.110	-13.9	0.903
9.2	0.681	10.945	8.076	-18.7	0.854
8.6	0.733	8.795	6.622	-22.4	0.835
8.0	0.785	5.936	4.203	-30.1	0.710

Run Number 31, PCG 612 1984 Jubail Fin Mod Acceptance Trial
FINS Active, Mod Installed, 15 Knots, 6-8 Ft Beam Sea
SKIP=3.5, Unfiltered FFT=7

9.8	0.638	2.035	2.634	-10.3	0.884
9.1	0.687	3.151	3.364	-11.8	0.918
8.5	0.736	5.610	5.265	-15.4	0.960
8.0	0.785	6.810	6.507	-11.8	0.931
7.5	0.834	12.635	13.276	-14.2	0.965
7.1	0.884	30.582	28.415	-16.5	0.978
6.7	0.933	49.034	45.761	-18.8	0.988
6.4	0.982	57.880	52.832	-17.6	0.988
6.1	1.031	33.830	32.510	-15.4	0.987
5.8	1.080	39.309	42.468	-17.9	0.994
5.6	1.127	50.252	50.953	-19.7	0.989
5.3	1.178	82.699	78.986	-21.4	0.993
5.1	1.227	81.791	76.0	-19.8	0.992
4.9	1.276	78.8	74.0	-20.5	0.988
4.7	1.325	57.4	53.2	-23.4	0.981
4.6	1.374	66.3	63.5	-24.6	0.988
4.4	1.424	74.2	70.6	-24.5	0.992
4.3	1.473	63.7	60.8	-24.3	0.989
4.1	1.522	86.2	85.5	-24.1	0.982
Peak →4.0	1.571	101.3	91.9	-27.4	0.984
3.9	1.620	70.2	60.3	-27.7	0.991
3.8	1.669	41.5	34.3	-29.1	0.985
3.4	1.865	29.6	27.9	-32.7	0.962
3.0	2.111	33.3	32.9	-35.7	0.956
2.6	2.405	11.8	12.6	-7.3	0.628

TABLE 5 - ROLL REDUCTION PERFORMANCE COMPARISON FOR PCG-612, WMEC-619 AND WMEC-901

15 Knots, 6 - 8 Ft Seas

Ship Class	Seaway	RMS Roll Unstabilized	RMS Roll Stabilized	Roll Reduction
PCG-612 (April 1984)	Bow	0.91°	0.71°	22%
	Beam	3.14°	1.46°	53%
	Quartering	2.85°	1.23°	56%
WMEC CONFIDENCE (February 1984)	Bow	--	--	--
	Beam	--	--	--
	Quartering	4.42	3.32	25%
WMEC BEAR (March 1984)	Bow	2.1	1.2	43%
	Beam	4.7	3.0	36%
		5.0	2.2	56%
	Quartering	6.2	5.0	19%
		5.5	5.9	-7%*

*19 knots in a decaying sea.

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